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TESTING OF NAVAL AVIATION GASOLINE.

By LIEUT. B. P. WARD, U.S.N., AND W. P. SINCLAIR.*

During the past five years, Naval Aeronautical engine operation has advanced with such rapidity that it has become astounding. This advancement has been due not only to the improvement in engine design and construction but to the use of improved gasolines. Formerly, engines were designed for operation with the available grades of gasolines, while now aviation gasolines are produced by the refiners to meet specified engine requirements.

Prior to 1927, gasoline for Naval Aeronautical use was purchased in accordance with the requirements of a specification which, in general, governed only the distillation range of the gasoline. Practically no requirements relative to the anti-knock and other

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properties which today are considered to be of great importance were contained in those specifications. Therefore, the gasolines so obtained were usually of low anti-knock value and the use of large quantities of expensive knock suppressing compounds in the fuel was required to produce satisfactory engine operation.

In order to obtain better grades and eventually more economical fuels, the Bureau of Aeronautics in 1927 arranged with the Bureau of Engineering for the U. S. Naval Engineering Experiment Station to conduct tests for the establishment of an acceptable list of aviation gasolines that would be suitable for Naval Aeronautical use. For this purpose fourteen different brands of aviation gasolines were submitted to the Experiment Station by various refiners.

The conduct of the test for the establishment of the acceptable list consisted of determining the physical properties and the anti-knock properties of the gasolines. The results of the test showed that most of the gasolines had satisfactory physical properties, but that there was a marked relative difference in their anti-knock properties. The better gasolines had Highest Useful Compression Ratios of approximately 5.5:1 or slightly greater. The gasolines having satisfactory physical properties and H.U.C.R. of 5.5:1 were considered to be acceptable for Naval Aeronautical use. On that basis nine of the fourteen gasolines were accepted. Using the properties of the acceptable gasolines as standard, a specification for aviation gasoline was prepared by the Bureau of Aeronautics. This specification was adopted and later became a part of the schedules issued by the Bureau of Supplies and Accounts prior to letting semi-annual contracts for aviation gasoline.

In order to meet the demand of the modern high compression engine and the requirements of the increasing range of Naval Aeronautical activities, frequent revision of the gasoline specifications was necessary. The revisions have been upward and so regular that gasolines which were found to be suitable for one contract six months later were found to be unsuitable for a new contract. Therefore, for each new contract suitability tests are conducted on samples of the gasolines which the prospective bidders intend to furnish.

The refiner who is a prospective bidder for a contract is required to submit a fifty-gallon sample of his gasoline to the Experiment Station and to defray the expense of testing it, which amounts to approximately $50.00. In return for the deposit, he is given a copy of the Station's complete test report. The report is confidential between the Bureau of Aeronautics and the refiner. It can not be used for advertising purposes. If the results of the test show the properties of the gasoline to be acceptable, the Bureau of Aeronautics places the refiner's bid under consideration.

The suitability test of gasoline consists of determining its physical and anti-knock properties and ascertaining if it conforms to the current requirements, which are as follows:

(a) Only gasoline consisting of a refined fraction of crude petroleum and/or with natural gas will be considered. Aromatic hydrocarbons may be added as specified below. The use of lead tetraethyl is prohibited.

(b) The gasoline shall be free from water and suspended matter.

(c) Anti-knock value. The anti-knock value shall not be less than 73 octane number and have a H.U.C.R. of 6:1 in an approved variable compression engine.

(d) Benzols and similar aromatic hydrocarbons may be added in such quantity that the blend meets all requirements for the finished fuel and the freezing point in particular.

(e) The color shall be 25+ Saybolt.

(f) Corrosion test. One hundred milliliters of the gasoline evaporated in a polished copper dish shall cause no gray or black corrosion.

(g) Gum—The gasoline shall be subject to an accelerated aging test at 212 degrees F. with oxygen at approximately 100 pounds per square inch pressure for four hours. On evaporation of 100 milliliters of oxidized sample (plus the addition of gum solvent), the amount of residue shall not exceed 10 milligrams.

(h) Distillation range. The limits are as follows: When the thermometer reads 167 degrees F., not less than 10 per cent shall be evaporated.

When the thermometer reads 212 degrees F., not less than 50 per cent shall be evaporated.

When the thermometer reads 275 degrees F., not less than 90 per cent shall be evaporated.

The residue shall not exceed 2 per cent.

The per cent evaporated shall be found by adding the distillation loss to the amount collected in the receiver at each specification point.

(i) Reid vapor pressure. The Reid vapor pressure limit at 100 degrees F. shall be 7 pounds per square inch maximum pressure. No tolerance will be allowed. (The pressure shall be determined as found in the appendix of the 1930 revision "Interstate Commerce Commission Regulations for the Transportation of Explosives.")

(j) Freezing point. The freezing point as indicated by the initial formation of solid rather than by the cloud point shall not be higher than minus 76 degrees F.

(k) Sulphur shall not exceed 0.10 per cent.

The tests for determining the anti-knock properties of gasoline in terms of Octane Numbers and Highest Useful Compression Ratio are conducted, respectively, in a "Series 30" Ethyl Gasoline Corporation knock testing apparatus and in a National Advisory Committee for Aeronautics Universal test engine.

The "Series 30" Ethyl Gasoline Corporation knock testing apparatus, as shown in Figure 1, is a commercial engine designed and built by the Ethyl Gasoline Corporation for knock testing of gasoline. It consists essentially of a single cylinder, 2½ inch bore, 4⅝ inch stroke, liquid cooled engine, with auxiliary equipment and a control panel. The engine of the apparatus is mounted on a cast iron base and is connected with pulleys and belt to a 2 horsepower synchronous motor generator mounted on the same base. A Midgley bouncing pin and its accessories are used for measuring the knock characteristics of the fuels in the combustion space of the engine. The pin element is screwed into a tapped hole in the head of the engine. This element, as shown in Figure 2, is a slender hollow cylinder which has a steel diaphragm locked in the combustion space end of it. The steel diaphragm supports a pin.

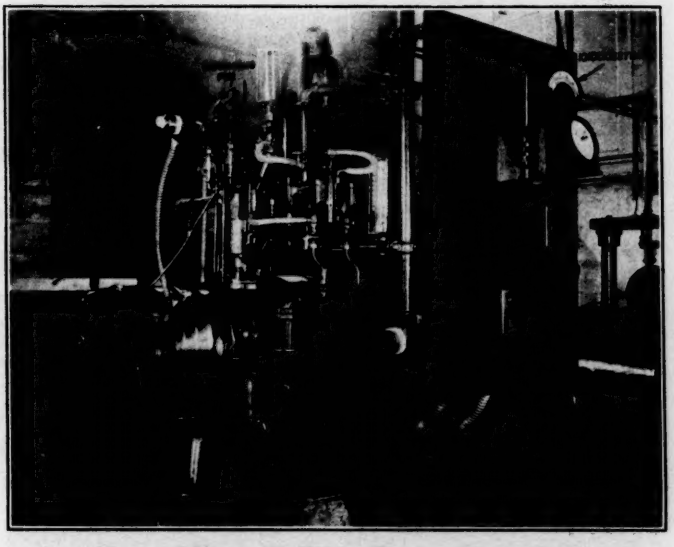

FIG. 1.—"SERIES 30" KNOCK TESTING APPARATUS.

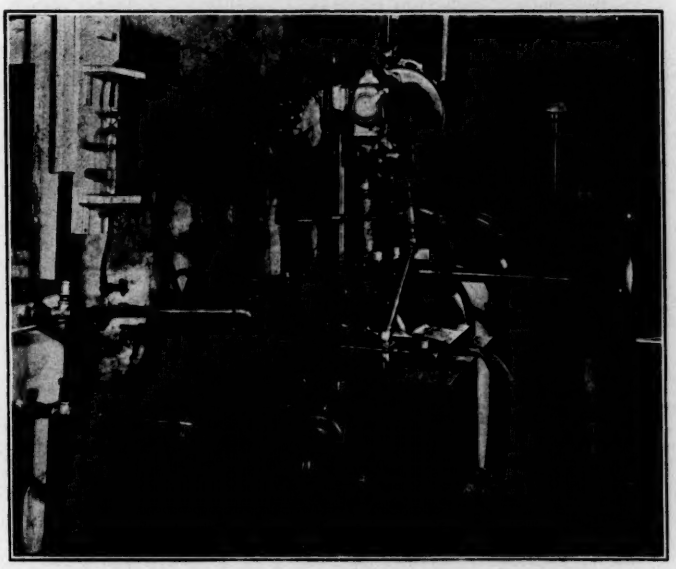

FIG. 4.—N. A. C. A. UNIVERSAL TEST ENGINE.

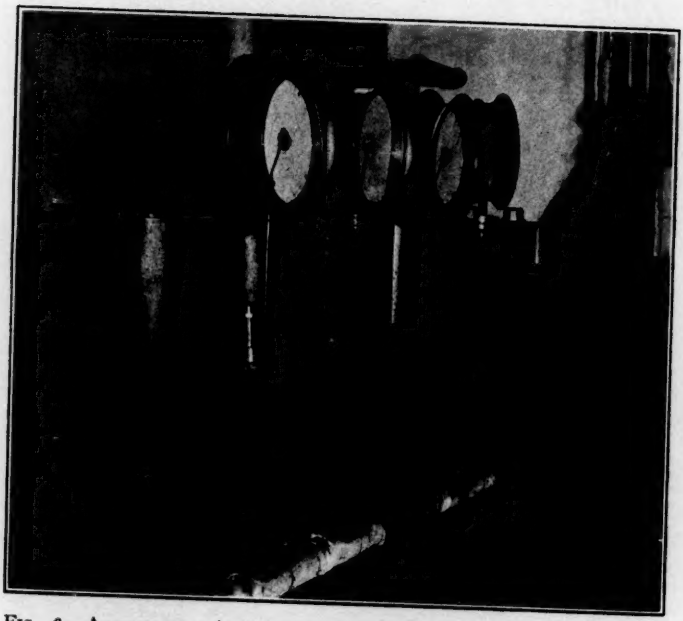

FIG. 6.—ACCELERATED AGING APPARATUS FOR GUM DETERMINATION IN GASOLINE.

which is free to bounce and close electrical contact points located on the top of the pin. Normal burning of the fuel in the engine produces a rate of pressure rise which is not sufficient to cause a sudden deflection of the diaphragm. Detonation or knocking, however, so alters the characteristics of combustion that the rate of pressure rise is tremendously increased and there is a consequent sudden deflection of the diaphragm which projects the bouncing pin upward to close the contact points. The higher the knocking properties of the gasoline, the greater will be the deflection of the diaphragm of the bouncing pin and the electrical contact points will remain closed for correspondingly longer periods of time. A knock-meter connected in the circuit with the contact points and mounted on the control panel of the apparatus, as shown in Figures 1 and 2, provides a means of measuring the relative knocking of fuels.

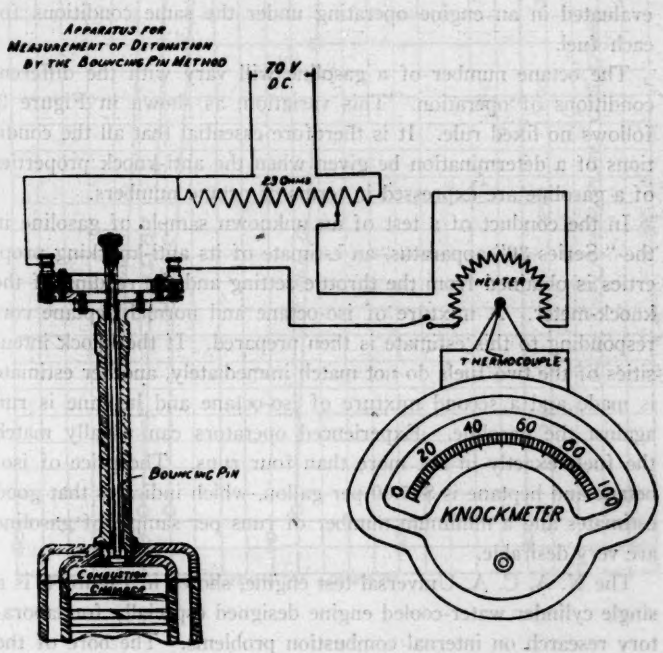

FIG. 2.—BOUNCING PIN CIRCUIT WITH KNOCK INDICATING MILLIVOLTMETER.

The procedure of the test consists of operating the engine under specified conditions and expressing the anti-knock properties in terms of equivalent blends of iso-octane and normal heptane. The current specifications require a constant engine speed of 900 R.P.M., spark timing set 16 degrees before top dead center, and a jacket temperature of 375 degrees F. The latter condition simulates the cylinder wall temperature of a medium compression ratio, air cooled engine operating at full throttle.

The octane number of a gasoline is the percentage of iso-octane used in a mixture of iso-octane and normal heptane which exactly matches the gasoline in intensity of knocking. Iso-octane and normal heptane are fundamental standard fuels having high and low anti-knock properties, respectively. If a gasoline has an octane number of 73, its anti-knock properties are equivalent to those of a mixture of 73 per cent iso-octane and 27 per cent normal heptane evaluated in an engine operating under the same conditions for each fuel.

The octane number of a gasoline will vary with the different conditions of operation. This variation, as shown in Figure 3, follows no fixed rule. It is therefore essential that all the conditions of a determination be given when the anti-knock properties of a gasoline are expressed in terms of octane numbers.

In the conduct of a test of an unknown sample of gasoline in the "Series 30" apparatus, an estimate of its anti-knocking properties is obtained from the throttle setting and the reading of the knock-meter. A mixture of iso-octane and normal heptane corresponding to this estimate is then prepared. If the knock intensities of the two fuels do not match immediately, another estimate is made and a second mixture of iso-octane and heptane is run against the gasoline. Experienced operators can usually match the fuels exactly in not more than four runs. The price of iso-octane and heptane is $25.00 per gallon, which indicates that good estimates and a minimum number of runs per sample of gasoline are very desirable.

The N. A. C. A. Universal test engine, shown in Figure 4, is a single cylinder water-cooled engine designed especially for laboratory research on internal combustion problems. The bore of the

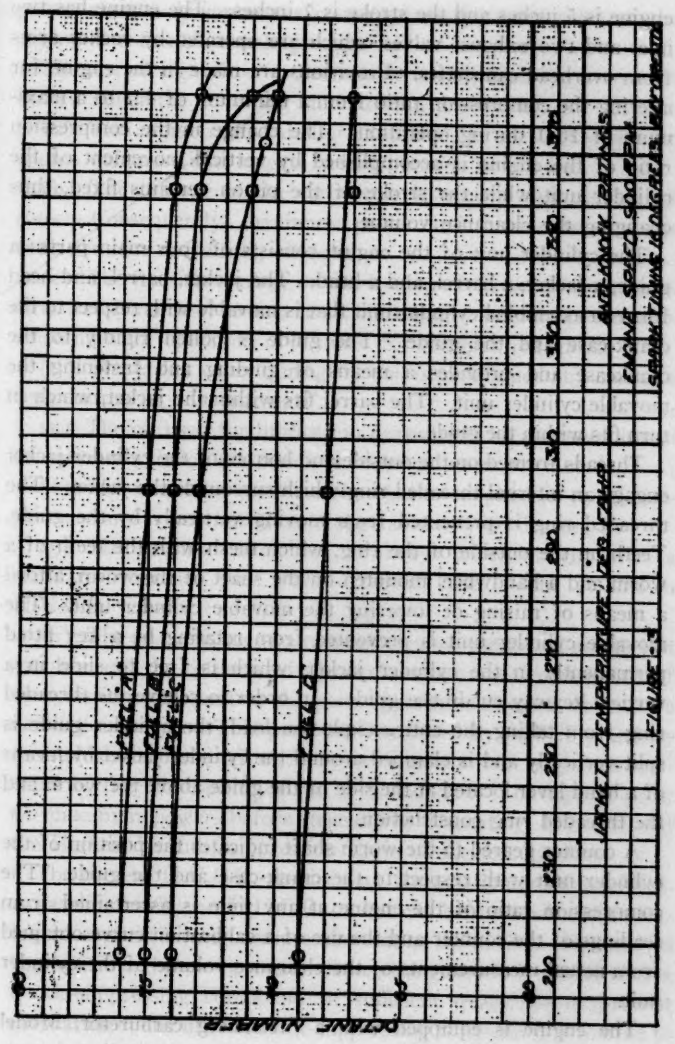
OCTANE NUMBER
FUEL A
FUEL B
FUEL C
MARKET TEMPERATURE DEG. F.AHR
ENGINE SPEED 900 R.P.M.
ANTI-KNOCK RATINGS
SERIES THREE IN ORDER IS RECOMMENDED

FIG. 3.

engine is 5 inches and the stroke is 7 inches. The engine has two inlet and two exhaust valves, which are operated by rocker arms from overhead camshafts. Provisions are made in the engine for altering the compression ratio from a minimum of 4:1 to a maximum of 15:1 during operation. The change in the compression ratio of the engine is accomplished by vertical movement of the cylinder unit while the stroke of the piston remains fixed, thus changing the clearance volume.

The cylinder unit of the engine consists of four main parts: a guide, a jacket, a barrel, and a head. The jacket, barrel, and head form an assembled cylinder unit that is movable with respect to the crank-case and the guide. The guide is bolted rigidly to the crankcase and provides a means of guiding and fastening the movable cylinder unit. The barrel fits within the jacket, which in turn fits within the guide.

Threads formed on the outside and bottom of the cylinder jacket engage an internal threaded ring which surrounds the jacket. The threaded ring is prevented from moving vertically by the guide. Teeth on the outside of the ring, which mesh with the teeth of a worm and a handwheel mounted on the shaft of the worm, afford a means of raising or lowering the movable cylinder unit. The movable cylinder unit is prevented from rotating by a key fitted permanently in the cylinder jacket, which is free to slide in a vertical keyway cut in the guide. In order to relieve the threaded ring from taking the entire explosion load, the cylinder guide is split vertically and is clamped around the cylinder jacket by means of a hand lever located at the side of the guide above the worm and the threaded ring construction.

A counter geared to the worm shaft indicates the position of the cylinder unit with respect to the crank-case and the guide. The compression ratio of the engine at any time is ascertained from readings of the counter and the use of a calibration curve obtained from actual measurements of the clearance volume of the cylinder unit.

The engine is equipped with a Stromberg carburetor, Model NA-S4, a Scintilla magneto, oil pump, and a water cooling system. The power developed by the engine is absorbed by a 50 horse-

power electric dynamometer connected to the main shaft of the engine by a flexible coupling.

The test of a gasoline in the variable compression engine is conducted under the following standardized procedure:

(a) The engine is operated at 1200 R.P.M., the best speed for engine quietness and longest period between overhauls.

(b) The throttle of the carburetor is opened wide during all runs. Consequently, maximum power and a constant fuel air ratio are obtained during all runs.

(c) The ignition timing, or spark advance, is set 25 degrees in advance of top dead center, at which setting maximum power with smoothness of operation is obtained for the specified speed.

(d) The jacket water during the run is maintained at a constant temperature of 180 degrees F.

(e) The oil used for lubrication purposes is an approved brand of summer aviation oil, which is renewed when warranted.

(f) The intake air to the carburetor is supplied at room temperature and is kept, usually, at approximately 80 degrees F.

(g) The gasoline under test is always tested directly with a standard reference fuel. The reference fuel is one of known H.U.C.R.

(h) The test consists of a number of runs conducted on the test gasoline and the reference fuel over a scale of different points of compression ratio and to the point in each case when destructive detonation occurs. This point is determined by audibility and by the sudden dropping off of the power output of the engine as indicated by the brake-beam of the dynamometer. The highest point of maximum power, before destructive detonating, is considered the Highest Useful Compression Ratio of the test gasoline.

The present H.U.C.R. requirement for aviation gasoline is 6:1. A gasoline having an H.U.C.R. of 6:1 can be used without knocking in engines having compression ratios up to 6:1. For use in engines of higher compression ratios, the gasoline is doped with a knock-suppressing compound in sufficient quantities to prevent detonation. The use of knock-suppressing compounds is limited to actual requirements.

The distillation range of a gasoline is a laboratory estimate of its volatility. A gasoline is required to possess sufficient volatility so that an explosive mixture is readily distributed to the cylinders for the starting and the operation of an engine. If the volatility of the gasoline is inadequate for this purpose, all the other advantages it may have are completely out-balanced.

A desirable range in the composition of a gasoline in regard to its volatility is one represented by a low initial boiling point, a fairly straight distillation curve, and a moderately high end-point. In order to estimate the volatility of a gasoline by distillation, at least three points should be known: The 10 per cent, the 50 per cent, and the 90 per cent points. These points, respectively, give an estimate of the properties of the gasoline for ease of starting, for acceleration, and for satisfactory operation.

The ease of keeping an engine running during the warming-up period depends somewhat upon the same characteristics of volatility as in the case of starting. The difference in these characteristics lies in the fact that the engine manifold and carburetor, during the warming-up period, are not at any constant temperature and the fuel-air ratio is less, particularly if a choke is used in starting. This means that a greater percentage of the fuel must be vaporized to form an explosive mixture after the engine has been started.

The method used for estimating the volatility of a gasoline by distillation consists of distilling at a given rate 100 cubic centimeters of the gasoline from a flask of specified dimensions. The flask contains a side arm through which the vapors pass to a condenser. The temperatures of the vapors leaving the flask are ascertained from a mercury-glass thermometer inserted in a cork stopper in the mouth of the flask. The apparatus used in the volatility determination is shown in Figure 5.

The color test, the corrosion test, and sulphur content requirements of a gasoline are primarily for the purpose of determining if the gasoline is properly refined. These properties and conditions of a gasoline have no effect on its main properties. A gasoline which in color is darker than 25+ Saybolt, (excluding added coloring agents), shows corrosive reaction, and contains more than 0.10

per cent sulphur, will corrode and cause destructive reactions on the internal parts of engines.

Some gasolines contain constituents which, during storage frequently oxidize and cause the formation of a gummy residue on evaporation of the gasoline. This condition is not ascertainable when the gasoline is fresh. Hence, an accelerated aging test for determining the gum formation has been devised. The test is conducted as follows:

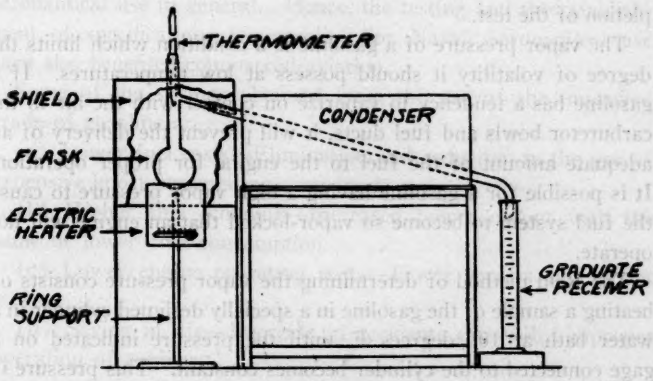

FIG. 5.—A. S. T. M. DISTILLATION APPARATUS.

An 8-ounce glass bottle, filled with gasoline to be tested, is covered with a loose glass top to prevent ingress of dirt. It is inclosed in the stainless steel bomb of the accelerated aging apparatus shown in the photograph, Figure 6. The bomb is then charged with oxygen at 100 pounds per square inch pressure. A water jacket surrounding the bomb is filled with fresh water, which is kept at the boiling temperature for the 4-hour period of the test. When oxidation of the gasoline begins, it is indicated on the recording gage connected to the bomb. Any appreciable diminution of pressure during the test shows that gum is forming in the gasoline. After the sample has been subjected to the conditions described above, it is removed from the bomb and analyzed for gum. The analysis is made by evaporating 100 milliliters of the

oxidized sample of gasoline to dryness in a glass dish over a steam bath. The dish is then transferred to an electric oven maintained at a temperature of 230 degrees F. for two hours; then it is placed for twenty minutes in a dessicator containing anhydrous calcium chloride as a drying agent. The dish is then weighed and the gum content is expressed in milligrams per 100 cubic centimeters of gasoline. Some gasolines will withstand the 4-hour accelerated aging test without the formation of gum, while others have shown that they contain as much as 500 milligrams on completion of the test.

The vapor pressure of a gasoline is a condition which limits the degree of volatility it should possess at low temperatures. If a gasoline has a tendency to vaporize on contact with the air in the carburetor bowls and fuel ducts, it will prevent the delivery of an adequate amount of the fuel to the engine for proper operation. It is possible for a gasoline having a high vapor pressure to cause the fuel system to become so vapor-locked that an engine will not operate.

The Reid method of determining the vapor pressure consists of heating a sample of the gasoline in a specially designed cylinder in a water bath at 100 degrees F. until the pressure indicated on a gage connected to the cylinder becomes constant. This pressure is then considered to be the vapor pressure of the sample.

The apparatus used in the vapor pressure test consists of a lower sampling cylinder, an upper vapor cylinder, and 0-15 pounds pressure gage. The lower sampling cylinder and the upper vapor cylinder are composed of two sections of 2 inch standard size pipe with closed ends, 3 and 10 inches long, respectively. These sections are connected together by means of two ½ inch × ¾ inch bushings and a ½ inch × 2 inch nipple.

The freezing point is an essential property of an aviation gasoline. In view of the low temperatures encountered at high altitudes, the freezing point of the gasoline should be low. Hence, a gasoline which shows no formation of solids at a temperature of minus 76 degrees F. is considered to have a satisfactory freezing point for aviation use.

In conclusion, it might be stated that the testing and the establishment of specifications for aviation gasoline have been a means

of obtaining for Naval Aeronautical use gasolines which are properly refined, which have high anti-knock properties, which have adequate volatility, and other required properties for satisfactory and successful operation of aviation engines.

The cost of these improved grades of gasolines has been about the same and even lower than the cost of much inferior grades of gasoline previously purchased. The conditions of the tests and the requirements of the specifications have been an incentive for refiners to produce better fuels, not only for Naval use, but for aeronautical use in general. Hence, the testing and the establishment of specifications for gasoline for Naval Aeronautical use have also benefited commercial aviation.

Some of the savings obtained from the use of the improved grade of gasoline are:

(a) Lower fuel costs—Elimination and reduction in the use of expensive knock suppressing compounds.

(b) Higher fuel economy—Increased power output with the same or lower fuel consumption.

(c) Lower engine operating costs—Fewer repairs and longer service.

(d) Saving in life—Decrease in accidents caused by improper operation of engines.

WAVES.

By Prof. Wm. H. Crew.*

PART I.

No one lives more intimately with waves than does the sailor! His life upon the bounding main has been duly recorded in history and his associations with the ocean's waves—"white sea-horses galloping o'er fenceless meadows"—have been justly sung and praised by the bards of all ages. The modern sailor, however, finds his life intricately involved with other kinds of waves;—waves not unlike those of the sea in their fundamental character, but quite different in their appeal to human senses.

For example, when the flag becomes fouled on its staff the quartermaster is sent aft to clear it. What does he do? By a sharp twitch of the hand a transverse disturbance or a solitary wave, if you will, is made to travel upward along the halliard and at the top it kicks loose the entangling folds of the flag. This wave is one of a type which we may genetically call "waves in ropes."

But again the sonorous chorus of bellowing fog-horns which greets the mariner as he runs up the Mersey River into the Port of Liverpool on a thick English morning is but sending its timely warnings in the form of waves or series of longitudinal disturbances *in the air*. These are called "sound waves." In recent years the sea-farer has been concerned with sound waves *in water*, particularly those of super-audible frequency,—the "supersonic waves," which the Grand Banks fisherman projects vertically into the sea to obtain the depth and the character of the bottom.

The great Cape Henry Light at the mouth of the Chesapeake and powerful San Juan Light on the eastern extremity of Porto Rico are typical sources of a third kind of wave which aids the sailor in piloting along the coast; the stars, which give him his "fix" at sea, are celestial light-houses billions of times more power-

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ful than those constructed by man. These aids to navigation emit *electromagnetic radiations,*—a ubiquitous family of waves of which *light* has long been a prominent member; although its popularity, particularly among sea-going men, is now critically threatened by *wireless waves,*—a recluse until recent years.

As different as they may be in their outward appearance and their appeal to man, all of these species of waves have striking similarities in their modes of motion and propagation, indicating a certain heritage common to them all. It is here proposed to discuss these waves with regard to their physical nature and to show how they are all related by a single fundamental equation; and finally to examine in somewhat more detail the various types of "water waves,"—the long waves, deep-water waves, the tides, and the ripples,—waves which are aesthetically the most fascinating, but scientifically the most difficult of comprehension.

It is important to obtain at the start a clear conception of wave motion, for the appearance of waves at sea are liable to lure one into believing that large masses of water are transported in the body of a wave. But one knows by observation of objects floating on water that such motion as actually exists is essentially oscillatory and that the water in any small portion of a wave moves but little from its position of equilibrium. What is it, then, that is whipped into motion when the wind blows over the sea?

It is the crest and the trough of a wave that travels horizontally;—or, in physics, one speaks of the *phase* as being propagated with such and such a velocity. All particles in the crests of a series of waves are said to be in the *same* phase, as is also said of those in the troughs; but those in the troughs are half a period out of phase with those in the crests. Thus, in general, one may say that a wave motion is an orderly assemblage of periodic motions so arranged in space and time that a given phase progresses in a definite direction and with a definite speed.

Waves in ropes: One of the simplest types of wave motion is the solitary wave in a rope. What experimental facts are known about such waves? Let us suppose we have two equal lengths (about 20 feet each) of rubber tubing stretched horizontally side by side. If these are simultaneously struck by the edge of the

hand single disturbances or waves will travel back and forth along each. By increasing the tension in one of the tubings, the wave in that one will be observed to travel with the greater velocity,—in fact the velocity will be doubled if the tension is quadrupled. It may also be observed that if the linear density of one of the tubes be increased, as is readily done by filling it with sand or a long spiral spring, the velocity of the wave in that one will be diminished. In short, this experiment when accurately performed indicates that *the velocity of a wave in a rope varies directly as the square root of the tension, and inversely as the square root of the density.*

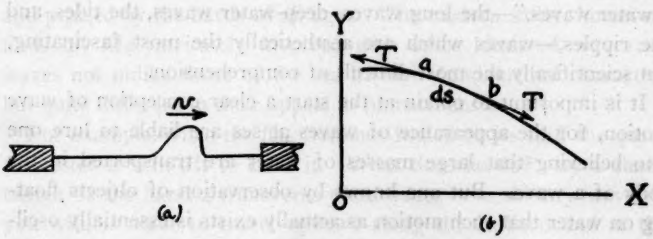

FIGURE 1.

Let us now discuss the solitary wave in a perfectly flexible rope, Figure 1-(a), with regard to its mass and motion,—that is, dynamically. If the displacement in the Y-direction, Figure 1-(b), is small, then the net downward force on any small length, ds, of the rope, due to the tension, T, is: $-T \frac{\partial^2 y}{\partial x^2} ds$. According to Newton's second law of motion this force may be equated to the mass-acceleration of that portion of the rope; thus, if ρ is its linear density, we have, after transposing certain terms and simplifying:

$$\frac{\partial^2 y}{\partial t^2} = \frac{T}{\rho} \frac{\partial^2 y}{\partial x^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Simon Newcombe, a distinguished professor in the United States Navy and at one time lecturer at Johns Hopkins University, is quoted as having once remarked to his students: "Gentlemen,

all the laws of nature are written in the form of differential equations." The above differential equation expresses the law governing the motion of a solitary wave in a rope. But we are thoroughly justified at this point in feeling a bit incredulous of this last statement, for such a stupid looking formula as is Equation 1, does not assist greatly in clarifying the situation.

To appreciate the significance of this *wave equation* one must integrate it,—a task first accomplished by the great French mathematician, d'Alembert, in 1747. His solution, after replacing the ratio $\frac{T}{\rho}$ by the symbol C^2, yielded the equation:

$$y = f_1(x - ct) + f_2(x + ct) \quad . \quad . \quad . \quad . \quad (2)$$

This equation gives us the displacement, y, of the rope at any position, x, along its length and at any time t. The actual value of y depends upon the form of the functions f_1 and f_2, and also upon the value of the constant, c.

That this equation represents two independent waves travelling in opposite directions along the rope, may be readily seen by giving y a constant value corresponding to the crest of a wave. Thus the first term of the right hand member of Equation 2 represents the crest travelling in the positive X-direction, for as time, t, increases, x must also increase to keep y constant; and the second term represents the crest travelling in the negative X-direction, for here x must decrease, with increase of t, in order to keep y constant. The constant c is the factor of proportionality between x and t; that is, it is the velocity of propagation of the crest along the X-axis.

Now this is an immensely interesting thing to know, for it means that the ratio, $\frac{T}{\rho}$, which we chose to represent by c^2, is really the square of the velocity of the wave; *i.e.*;

$$v = \sqrt{\frac{T}{\rho}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This result is in agreement with the experimental facts noted on page 300; namely, that the velocity of propagation of a wave in

a rope varies directly as the square root of the tension of the rope and inversely as the square root of its density.

We may now view more intelligently the differential wave equation (Equation 1), if we write it:

$$c^2 = \frac{\frac{\partial^2 y}{\partial t^2}}{\frac{\partial^2 y}{\partial x^2}} = (\text{velocity})^2 = \frac{\text{acceleration}}{\text{curvature}} \quad . \quad (4)^*$$

In words this means that the square of the velocity of propagation of a wave is equal to the ratio of the acceleration of a particle of the medium at any point to the curvature at that point of the x-y curve; *i.e.*, the curve of the instantaneous wave form. It is important to bear in mind that this interpretation of the wave equation holds only for those cases in which the motion of the medium is in a plane normal to the direction of propagation.†

As yet nothing has been said about the nature of the functions f_1 and f_2 in Equation 2. They are perfectly arbitrary and are chosen so as to represent the form or shape of the wave. To actually produce a *series of waves* in a rope one must agitate it repeatedly; that is, disturb it periodically. Consequently, to represent such a wave train by Equation 2 one must choose periodic functions of x and t.

For example, the wave train, shown in Figure 2, moving with velocity v in the positive X-direction may be represented by:

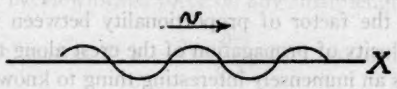

FIGURE 2.

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* Actually the term $\frac{\partial^2 y}{\partial x^2}$ is the "flexion," *i.e.*, the space rate of change of slope;—but this is very approximately the curvature, which is: $\frac{\partial^2 y}{\partial x^2}\left[1 + \left(\frac{\partial y}{\partial x}\right)^2\right]^{3/2}$ See Davis, "The Calculus," page 169 [Ed. 1913].

† In the case of the rope this condition was imposed when the acceleration was represented by $\frac{\partial^2 y}{\partial t^2}$.

$$y = f(x - vt) = A \cos a (x - vt) \quad (5)$$

and is commonly known as a *progressive wave.* Two progressive waves of this type moving with the same velocity but in opposite directions combine to produce a *standing wave,* having the characteristic loops and nodes which are easily seen in a vibrating violin string. It is a matter of but little dexterity to produce standing waves in a rope or a rubber tubing fixed at one end by carefully shaking the other so that the departing waves meet those reflected from the fixed end.

What we have done so far is to consider the dynamics of a solitary wave in a perfectly flexible rope and to derive thereby a very general wave equation; so general, indeed, that it may represent a solitary wave travelling either to right or left, or both at once, a series of waves, or two series opposing one another so as to render standing waves; and for all of these the velocity is completely determined by the tension and density of the medium. It will be interesting to see that the wave equations for other kinds of waves are of this same form and generality.

Sound Waves: Air is the medium by which sound usually reaches the ear. In considering waves in air one must recall a very fundamental distinction between solids and fluids; a distinction which is clear from the following observation: If a piece of rope is subjected to a small *shearing stress* in two adjacent planes, it will experience a slight strain and will therefore return to its normal condition upon removing the stress. No such strain, however, is set up in a similar section of water or air when a shearing stress is applied; so that if once distorted there are no internal forces tending to restore the fluid to its initial shape. But for *compressional stresses,* both the solid and the fluid are strained, and each experiences restoring forces. Therefore, solids are said to have coefficients of compressional and shearing elasticity, but fluids only those of compression.

In fluids, therefore, one cannot produce waves having displacements of the medium perpendicular to the direction of propagation as was the case with the rope; the displacements of waves in air are longitudinal and are called *compressional waves.*

Let us now study the dynamics of a plane wave of sound progressing in the X-direction along the axis of a column of air of

unit length and unit cross-section. If, by some mechanical means such as clapping the hands, the pressure on this unit volume is increased by an amount dp, then the volume will be decreased by an amount dv, the factor of proportionality between the two being the compressional coefficient of elasticity, E. Mathematically this statement is written: $dp = -E\,dv$. But the change in the volume of the column of air depends upon the difference in the amount that the two ends of the column are shifted from their position of equilibrium; therefore, if we denote the displacement of the medium in the X-direction by ξ, then $\frac{\partial \xi}{\partial x}$ is the differential displacement in the unit length, and we have $dp = -E \frac{\partial \xi}{\partial x}$

This increase of pressure is not uniform along the length of the column for it requires a finite time to travel from one end to the other; hence the difference of pressure on the two ends is $\partial p = \frac{\partial (dp)}{\partial x}$, and this *excess of pressure* on the anterior face will cause the included mass, ρ, of air to experience an acceleration $\frac{\partial^2 \xi}{\partial t^2}$. Writing this in the form of Newton's second law of motion:

$$-\partial p = \rho \frac{\partial^2 \xi}{\partial t^2} = -\frac{\partial}{\partial x}\left[-E\left(\frac{\partial \xi}{\partial x}\right)\right]$$

which reduces to:

$$\frac{\partial^2 \xi}{\partial t^2} = \frac{E}{\rho} \frac{\partial^2 \xi}{\partial x^2} \quad \ldots\ldots\ldots \quad (5)$$

One recognizes immediately the similarity of this and Equation 1, and can write down, by analogy, the expression for the velocity of sound waves; thus:

$$v = \sqrt{\frac{E}{\rho}} \quad \ldots\ldots\ldots \quad (6)$$

Now if the compressions and expansions of a mass of air take place at constant temperature, it is a direct result of Boyle's Law that $E = p$; whence, therefore, *the velocity of sound varies directly as the square root of the pressure and inversely as the square root of the density of the air.* Newton was the first to employ this expression in calculating the speed of sound but obtained a value, 920 feet per second, which was somewhat lower than the best experimental value of his time. This discrepancy he endeavored to explain by supposing that sound actually travelled through the solid substance of the air molecules with a velocity which was essentially infinite, and through the intervening space with the speed determined by him.

The real explanation awaited the consideration of Laplace in 1816, when he realized that the compressional waves in air are so rapid and the air itself so poor a conductor of heat that the processes of compression and rarefaction are essentially adiabatic. Consequently one must employ the adiabatic gas law which gives: $E = \gamma p$, where γ is the ratio of the specific heats of air. We have, then, for Laplace's expression for velocity:

$$v = \sqrt{\frac{\gamma p}{\rho}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

This gives a value for v of 1087 feet per second, which is in good agreement with experiment.

It is a very pretty problem to solve Equation 5 for the case of standing waves in air and an ample and interesting treatment of this is given in Horace Lamb's *Dynamical Theory of Sound;* but for the sake of brevity it must suffice here merely to see the striking analogy between the wave equation for waves in ropes and for sound waves. It is perhaps needless to say that the foregoing discussion applies not only to sound waves in air, but equally as well to the supersonic waves in water which may be excited by large piezo-electric oscillators.*

Electromagnetic Waves: It is quite impossible in this very general discussion to adequately treat the electromagnetic waves. Such neglect of them as it may be necessary to adopt should not be con-

* "Concerning the Piezo-Electric Effect"; U. S. Naval Institute Proceedings; 56; 805 (1930).

sidered as an indication of a lack of importance of these waves which scurry at the rate of 186,000 miles per second to the most remote corners of the universe, which inundate all space with their disturbances, and which travel for millions of years, along many millions of millions of miles, that on a few brief occasions an infinitesimal portion of them may be captured in the astronomer's telescope and advise him of a distant island universe mightier than the one he calls his own.

Nor should one minimize the far reaching influence of the other waves of the great electromagnetic spectrum; radio waves, which bring symphony concerts into our very parlors; the infra-red waves, bearing warmth from the sun; ultra-violet waves, which give us health; the X-ray and γ-ray waves, important tools in the science of medicine and surgery; and even the cosmic rays, which are still rather a mystery to man. We shall be satisfied to discover the general nature of an electromagnetic wave, to give its wave equation, and to observe a few consequences of this expression.

Following the firm establishment of Huygen's Wave Principle by the famous interference experiment of Thomas Young (1783-1829), efforts were made to establish a wave theory of light. The important facts to be satisfied by any theory are those of reflection, refraction, interference, and polarization. An early point of view was that the luminiferous ether was an elastic solid, having both the elasticity of compression and of shear. But this became untenable when Fresnel clearly showed from facts of polarization that the displacements in a light wave are transverse only. It has already been mentioned that fluids maintain longitudinal displacements, but no substance is known which transmits *only* transverse displacemets. Thus, in postulating the ether, we introduce a hypothetical substance which is unknown in actual experience in that it possesses only elasticity of shear.

In the year 1864 a notable British physicist and mathematician, Clerk Maxwell, propounded the famous electromagnetic theory, in which his chief contribution was the idea of displacement currents in a dielectric. Now these currents, established by the charging of a conductor, set up magnetic fields in the surrounding

dielectric. These fields, in the process of being established, induce secondary currents, which, in turn, induce secondary fields, and so forth, one current producing a field and the field, a current; the process progressing continuously out into space or through the dielectric.

Maxwell described these ideas in a set of differential equations, relating electric intensity, E; the magnetic intensity, H; the dielectric constant, k; and the magnetic permeability, μ. When applied to the case of the *plane wave*,* the equations of Maxwell reduce to the following:

$$\left.\begin{aligned} \frac{\partial^2 H}{\partial t^2} &= \frac{1}{k\mu} \cdot \frac{\partial^2 H}{\partial x^2} \\ \frac{\partial^2 E}{\partial t^2} &= \frac{1}{k\mu} \cdot \frac{\partial^2 E}{\partial x^2} \end{aligned}\right\} \quad . \quad . \quad . \quad . \quad (8)$$

From the previous study of waves in ropes, we are able to see that these equations represent two wave disturbances, one magnetic and one electric, being propagated along the X-axis with a velocity, $\sqrt{\frac{1}{k\mu}}$. It may be added that the electric and magnetic disturbances are at right angles to one another and each transverse to the direction of propagation.

The constants k and μ can be determined independently by electrical experiments, and when introduced into the above expression for the velocity of an electromagnetic wave yield the well-known value of the velocity of light. It was this fact that first led Maxwell to suspect that light was actually of an electromagnetic nature; but subsequent investigations disclosed the fact that other types of radiations, from wireless waves to cosmic rays, partake of an electromagnetic wave motion.

PART II.

Water Waves: It has already been mentioned that sound waves are propagated through water; but waves which one sees on oceans, lakes, and ponds are of a distinctly different kind. In the former the displacements are longitudinal; in the latter they are,

* Starling: "Electricity and Magnetism," Chapter XIV.

in general, elliptical. Also, in sound waves the restoring forces were of the nature of elastic cohesions and repulsions of the molecules, one for another; in water waves they result from two causes: gravity and surface tension. In the discussion that is to follow it will be seen that gravity waves are the ones commonly observed at sea and are called *deep-water waves* or *long waves* according to whether the ratio of the length of wave to depth of water is a small or large quantity. The *tides* are, perhaps, an anomolous example of long waves; or, more simply, a collection of oscillating waves. Waves in which surface tension provides the restoring force are commonly called *ripples*.

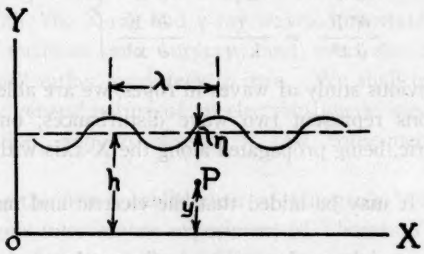

FIGURE 3.

For the sake of simplicity, let us consider waves in a rectangular tank of water of unit breadth, infinite length, and mean depth, h. We shall assume firstly, that the height of the surface of a wave above mean level, denoted by η, is small compared with the depth; secondly, that the density of the water, ρ, is constant; thirdly, that because of the incompressibility of the water and the proximity of the bottom the water particles are constrained to move in horizontal directions only, and also that all particles in any plane normal to the length, or X-axis, partake of their motions in unison. All of these assumptions conform closely with the actual *long waves*.

It is now possible to set up the following three equations:

$$\frac{\partial p}{\partial x} = g\rho \frac{\partial \eta}{\partial x} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$h.\,\delta x = \left(h + \eta\right)\left(1 + \frac{\partial \xi}{\partial x}\right)\delta x \quad . \quad . \quad . \quad (10)$$

$$\rho . h . \delta x . \frac{\partial^2 \xi}{\partial t^2} = - \frac{\partial p}{\partial x} . \delta x . (h + \eta) \quad . \quad . \quad (11)$$

Equation 9 is an *equation of motion.* It indicates that the variations in pressure, p, at any point, with respect to distance along the X-axis are proportional to the corresponding variations of the height of the surface, η, above that point; and the factor of proportionality is $g\rho$, g being the value of acceleration of gravity and ρ the density of the water.

Equation 10 is an *equation of continuity,* which equates the volume of a quantity of the water when undisturbed to the volume of the same quantity when partaking of a wave motion. These volumes are equal because of the incompressibility and continuity of the water. The symbol ξ denotes the horizontal displacement of a particle.

Equation 11 equates mass-acceleration to the force acting, in accordance with Newton's second law of motion; the minus sign being due to the fact that an increase in pressure along the X-axis gives an acceleration of the mass in the opposite direction.

If now we differentiate Equation 10 with respect to x alone and solve it for $\frac{\partial \eta}{\partial x}$ it may be combined with Equation 9 to render a simple value of $\frac{\partial p}{\partial x}$. This latter expression may be substituted in Equation 11 to give the following:

$$\frac{\partial^2 \xi}{\partial t^2} = g h \frac{\partial^2 \xi}{\partial x^2} \left(1 + \frac{\eta}{h}\right)^3 \quad . \quad . \quad . \quad . \quad . \quad (12)$$

But by our first assumption, $\frac{\eta}{h}$ is small; therefore neglecting this we have the familiar wave equation:

$$\frac{\partial^2 \xi}{\partial t^2} = g h \frac{\partial^2 \xi}{\partial x^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

for which the wave velocity is: $v = \sqrt{g h}$; *i.e.,* this velocity varies only with the depth (assuming that g is constant). One might have expected to find here some dependency upon the density of

the medium, as was the case with ropes and sound; but here we are dealing with waves sustained by gravity, under the action of which bodies move independently of their masses.

One who has been to the sea-shore may have wondered why waves which are approaching the shore obliquely invariably wheel around so as to render their crests more nearly parallel to the beach. The explanation is readily found in the velocity expression for these long waves; for the inshore end of a wave is travelling in less depth of water than the seaward end, and is therefore moving with less velocity, so that its front is refracted toward the beach.

Another interesting phenomenon of the sea-shore is the breaking of waves. The beach at Waikiki is well known for its huge and long sustained breakers. We may look again at the velocity expression for the explanation, for as a wave advances upon a shelving beach its rear part is ever in deeper water than is its forward part; consequently the rear will gradually overtake the front, causing the wave to mount higher and higher, until finally the crest topples forward in the form of a breaker. It is well to bear in mind that the initial assumptions made on page 308 have rendered this theory of the breakers very simple, and that actually the situation is much more complex. However, the simple wave equation is the firm basis on which are built all further refinements of the theory of long waves.

If, instead of restricting the motions of particles in a wave to the horizontal, we assume the more general elliptical motion, then one may deduce with rather more involved mathematical calculations* the following expression for the velocity of a water wave:

$$v^2 = \left(\frac{g\,\lambda}{2\,\pi} + \frac{2\,\pi\,T}{\rho\,\lambda}\right) \tanh \frac{2\,\pi\,h}{\lambda} \quad . \quad . \quad (14)$$

where λ is the length of the wave, h the depth, and T the surface tension of the water; a factor, operative in wave motion, which was first recognized by Stokes in 1848. For the *long waves,* which we have just discussed, the ratio, $\frac{h}{\lambda}$ is small and the hyperbolic tangent term in Equation 14 then reduces approximately to the

* Horace Lamb: "Treatise on the Motion of Fluids."

argument of the function itself; thus dropping further terms involving $\frac{h}{\lambda^2}$, which are small to the second degree, one is left with $v = \sqrt{g\,h}$, which is identically the previous expression for velocity of long waves.

In the case of waves for which the ratio $\frac{h}{\lambda}$ is large,—these are called *deep-water waves,*—the hyperbolic tangent term in Equation 14 becomes approximately unity, leaving the expression:

$$v = \sqrt{\frac{g\,\lambda}{2\,\pi} + \frac{2\,\pi\,T}{\rho\,\lambda}} \quad . \quad . \quad . \quad . \quad . \quad (15)$$

If the waves are comparatively long, *i.e.,* λ is large, then one may neglect the second term under the radical, in which case the velocity of the waves varies directly as the square root of their length. On this basis, calculations show that an Atlantic roller measuring 300 feet from crest to crest will travel at the rate of 27 miles per hour. It may further be shown that though such a wave may have a difference *in height* from crest to trough of 40 feet, still at 300 feet below the surface the displacement of a particle of water from its mean position is less than half an inch. Thus a mid-Atlantic gale merely ruffles the surface of the ocean.

For waves of very small length, such as those commonly known as *ripples,* the second term of Equation 15 becomes the relatively important one and the velocity is then seen to be *inversely* proportional to the square root of the wave length. Since, then, deep-water waves travel more slowly and ripples more rapidly as their lengths become less, there is obviously a critical wave-length which separates waves of the first type from those of the second. This critical water wave, for which the velocity is a minimum, is about three-quarters of an inch in length.

Of especial interest to the sailor are the waves produced by ships; and, indeed such waves are a matter of particular concern to the designer and constructor of vessels.

About one hundred years ago a curious observation was made on a canal in Scotland. A horse drawing a barge along the canal

became frightened and ran away, and it was evident to the driver, after the barge attained a certain high speed, that then much less effort on the part of the horse was required than at a somewhat lower velocity. This apparent paradox was studied a few years later by J. Scott Russell, an eminent British engineer, in connection with the institution of steam barges on the canals. He found that at a certain critical speed the barge produced but a single wave which travelled along with the barge, bearing it on its crest with bow and stern partly out of water. This "solitary wave," as Russell called it, is merely an example of a long wave, and the critical speed of the barge is the natural velocity of the wave corresponding to the depth of water in the canal. At speeds greater or less than this value, trains of waves follow in the wake of the barge; and the additional effort required to draw it at less than the critical speed is used up in maintaining such a train of waves.

These facts are well recognized in the making of trial speed runs of large ships. A very striking example * is the trial run of a British destroyer where, in a 7½ fathom course on the Maplins, 9000 horsepower gave a speed of 22 knots; whereas on the 40 fathom course at Skelmorlie, near the mouth of the Clyde, the same power propelled the ship at 26 knots. But at double that power the speed on the Maplins was 34½ knots, and at Skelmorlie only 33 knots. Apparently these latter speeds were close to the critical value for the 7½ fathom course, but still remote from that of the 40 fathom course. It is also interesting to note that pilots of sea-planes find that the necessary speed for "taking off" is attained more readily in shallow water than in deep; a fact which appears to conform with the curious observation made on the canal in Scotland.

The expenditure of power in propelling a ship goes into overcoming frictional and eddy losses, and in establishing various groups of waves. This second type of power loss has been given considerable study, particularly in England, by Mr. W. Froude. A large towing tank, in which is studied the waves from models molded in paraffin, is operated at the National Physical Laboratory at Teddington (near London), and smaller ones are in use in the United States at Washington and Ann Arbor.

* E. L. Attwood: "The Modern Warship," page 128.

Photograph from "Wide World Photos."

As may be clearly seen in the accompanying photograph of the Yale-Harvard crew race there are two sets of waves produced by a ship moving through water, one at the bow, the other at the stern. Each of these sets consists of a series of echelon waves diverging from the course of the ship with crests sloping aft, and a transverse series with crests nearly perpendicular to the course.* The formation of the echelon waves exacts energy from the ship, but they immediately pass away from her side into quiet water. The transverse waves, on the other hand, travel along with the ship; and, like true deep-water waves, they adjust their length to conform with their velocity in accordance with Equation 15; so that the faster a ship goes, the longer are these tranverse waves.

At a certain speed the length of the wave, λ, becomes equal to the length of the ship, L; that is, neglecting the second term of Equation 15, $v = \sqrt{\frac{g\,L}{2\,\pi}}$; or, if the velocity is expressed in knots, this becomes: $\frac{V}{\sqrt{L}} = 1.33$. Now it is an established fact that a vessel experiences a large increase of wave resistance when the crests of the transverse bow-wave system coincides with those of the transverse stern-wave system. This occurs when the length of the transverse waves is equal to the length of the vessel; *i.e.* when the ratio of the speed of the vessel to the square-root of its length is equal numerically to 1.33. For example, the length of a 15-knot deep-water wave is about 130 feet, so that it is quite uneconomical to design a vessel of this length to operate at 15 knots; whereas, quite the contrary would be the case for a 500-foot ship. Atlantic liners and battleships are built with a ratio of about 0.9; but high speed cruisers have a slightly higher ratio of about 1.1.

Ocean tides, which are observed as a periodic rise and fall of the sea-level along any coast, have been the subject of varying theoretical speculations for many years and have evaded a completely satisfactory explanation up to the present day. A most

* Encyclopaedia Britannica 9th edition article: "Shipbuilding."

commendable discussion of these mammoth waves is to be found in "The Tides," a small book by H. A. Marmer, and in an article by the same author published in the *United States Naval Institute Proceedings* for November, 1930. The explanation endorsed by Mr. Marmer considers the oceans as a mosaic of oscillating basins of water actuated, of course, by the combined actions of moon and sun. The magnitude, extent, and periodicity of these oscillations is also dependent upon the depth, length, and breadth of these basins. These oscillations of the water partake of what is called a stationary wave motion, and are analogous to the "standing waves" in a rope mentioned (page 303) in the first part of this article.

Although the progressive-wave theory of the tides is not so satisfactory as the foregoing one, it might nevertheless be in order to mention it here as it provides a very beautiful example of a most interesting physical phenomenon. In this discussion we may take an ideal situation in which two gigantic waves separated by half the circumference of the earth progress in a westerly direction with a speed of about 1000 miles per hour. Since the average depth of the sea is somewhat less than three miles, these waves—the tides—are of the type that we have previousy designated "long waves," for which the *natural* velocity (page 309) would be roughly 500 miles per hour. Actually, however, by the combined action of sun and moon, they are forced to travel at about 1000 miles per hour; *i.e.* at twice their natural speed. These are generally called "forced waves." If now we digress just a moment to a consideration of forced vibrations in a pendulum we shall be able to see by analogy the dynamical reason for the long known fact that in the unobstructed ocean *low* tide occurs directly under the moon just where one would expect the forces of attraction to produce *high* tide.

If a walking stick is inverted and held lightly between the thumb and forefinger, it will swing freely as a pendulum with a time of vibration called its "natural period." It may be forced into a vibration of a period longer than its natural one by moving the supporting hand to and fro through a small arc *in phase* with the swing of the stick. Or, it may be forced into a period shorter

than its natural one by a similar motion of the supporting hand, but this time with a *phase just opposite* to that of the stick; *i.e.* as the stick swings left one moves his hand to the right, and vice versa. In short, *if a body is forced to vibrate with a shorter period than its natural one, the driving force must be opposite in phase to the motion of the body.*

This is precisely the case of the tides; the moon is forcing a mammoth wave to circumvent the earth with a period less than its natural one, and consequently the moon and tide are opposite in phase. Actually, of course, this ideal situation is distorted by many factors—such as, for example, the non-uniform depth of the sea, the rotation of the earth, the effect of continents, influence of the sun, etc., etc.

The beauty in water waves is as varied as the types which have just been discussed; from the fine, sharp motion of ripples produced by pitching a pebble into a pond to the huge waves that break upon a shelving beach there are many fascinating physical features. The regular array of waves from boats displays phenomena which have their analogues in the realm of optics,—for example, referring again to the photograph of the crew race one may see near the center the *interference fringes,* that is, the linear loci of places where the troughs of one set of waves annul the crests of another; at the top center one may see *diffraction* or *secondary wavelets* produced at the small craft anchored on the right of the course; at the upper right there is *refraction* or *bending of waves* due to varying depth of the water. With regard to the tides, the true might and magnitude of these mammoth waves is perhaps no more elegantly realized than when one stands upon the battlements of Mont St. Michel and observes the edge of the sea advancing at the pace of a rapid walk, coming up from a distance of several miles to completely surround the historic mount which watches over the Normandy and Brittany coasts.

Thus, with the advance of knowledge in the field of light, sound, and electricity, and the applications of this knowledge to maritime pursuits, the sailor of today finds himself intimately associated with waves in water, air and ether. In the light of very recent discovery in physics one might extend the undulatory inti-

macy to still another realm—that of matter, both animate and inanimate, gaseous, liquid, or solid; for within the last decade it has been shown that the two minute physical entities known as electrons and protons behave as if each had associated with it tiny groups of waves*; and since it is well recognized that all matter is constructed entirely of these two constituents, with the concept of energy serving as a kind of mortar, one may consider that material things, earth, air, fire, and water—even our bodies—are but a manifestation of the existence of a curious mosaic of groups of waves. But let us not tarnish the romantic luster of the life at sea with these acrimonious differential equations and etherial speculations; rather let us sing praise to the glorious past and hope for the advancing future of the men who "go down to the sea in ships."

* Davisson and Germer of New York have shown that electrons reflected from single crystals of nickel obey the laws of the wave theory of light. A. J. Dempster of Chicago has shown a similar behavior of protons.

THE TREND IN NAVAL ENGINEERING.

Oil Purifiers, Laundry Washers, Steering Controls, Voice Projectors, and Searchlights.

By C. Huey, Civil Member.*

Note by Editor:—This is the eighth of a series of articles on "The Trend in Naval Engineering," the first of which was published in the February, 1931, number of the Journal.

This article continues the previous articles which discussed the principal functions and trend in design of various accessories or devices used in Naval service which have served to help in the evolution to higher efficiencies, quicker results, saving in time, saving in weight or saving in man-power.

OIL PURIFIERS.

The use of lubricating oil in Naval service has brought with it the same problem as experienced in commercial marine service, that is, the necessity of supplying larger amounts of clean lubricating oil commensurate with the increase in horsepower and machinery speeds.

With the passing of the older methods of lubrication employing drip, wick or manual oiling of the bearings, gears or cylinders, and the substitution of circulating and forced feed systems involving large quantities of oil in active service, the re-use of the same oil became an economic necessity.

It is a generally accepted fact that once a lubricant has proved satisfactory for a certain service, its subsequent failure to perform properly is not the result of internal change, but of contamination with foreign materials or the formation of sludge. The foreign materials usually consist of scale and rust, metal particles abraided from the bearings, atmospheric dust, core sand from castings, paint

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flakes, and water from atmospheric condensation, steam leakage or cooling water leakage.

The sludge is formed in the oil under the adverse conditions which the oil must meet. It is a heavy, muddy formation of oxyhydrocarbon caused by oxidation of the oil. While the usual form of sludge is not particularly abrasive, it is highly undesirable for the reason that it coats cooler tubes, prevents heat transfer, clogs the pipes and orifices of the lubricating system, and, when water is present, aids in forming emulsions. The effect of emulsions is to rob the bearing surfaces of necessary oil and to carry along foreign matter which would otherwise tend to settle out.

The importance of keeping the oil in circulatory lubricating systems free from impurities classified in general as foreign materials, sludge and water, cannot be over-estimated. With the use of an efficient purifying system, there is a distinct saving in the cost and stowage of replacement oil. From the operating standpoint there is a lowering in friction losses, reduction in bearing wear, and less frequent repairs.

The methods in use for the purifying of lubricating oil in the earlier days, employed gravity filtering or gravity settling tanks (see Plate No. 1). The objection to the *filtering* systems, of course, is the extreme slowness of operation. It is obvious that the quantity of oil which will pass through a filter, decreases with the effectiveness of the filtering material. Clogging of the filtering material to the point of complete inoperation is certain. With the gravity *settling* system, although large areas of settling trays are employed, the speed of purifying is dependent on the small force of gravity, and the appreciable distance through which the separated materials move to acquire separation. Considerable time and the avoidance of agitation is required for this form of purifying, and due to the small forces employed on materials of only very slight difference in specific gravity, the separation is never a sharply defined line between the purified oil and the foreign materials. Continuous flow of oil through gravity systems of purifying is impracticable, which aspect alone renders them unsuitable for modern machinery installation.

The modern high speed, *continuous* flow system of purifying oil or other liquids employing *centrifugal* force, is shown in Plate No. 2. In this system the oil is conducted downward through the hollow center of a rotating member or bowl, which is equipped with a series of closely spaced conical shaped discs. The oil passes upward from the bottom of the bowl through several series of holes in the discs and is distributed in thin layers between them.

The spacing of the discs is made as small as practical for the class of liquids to be purified, so that the movement of the foreign material from the thin film to the undersurface of the individual discs is as short as possible. The effect of centrifugal force on the foreign material, such as water, sludge, solid matter, etc., which are all heavier than oil, causes these materials to be thrown outward from the oil film against the underside of a disc from which the particles slide off into the sediment-holding space at the periphery of the bowl. The oil moves inward and upward to the discharge point. The separated water and lighter foreign particles are forced to the water outlet of the bowl by displacement, while the heavier foreign materials are retained at the outer wall of the bowl from which they are eventually removed at periodic cleaning times.

The rotating member of the centrifugal purifier described is revolved by means of a steam turbine wheel directly connected to the vertical spindle which carries the bowl, or by an electric motor belted and geared to this spindle in such a way as to obtain the necessary multiplication of speed to drive the bowl at about 6500 R.P.M. At this speed, the extreme centrifugal force exerted in the bowl is several thousand times that of gravity, and this, together with the fact that the ratio of the length of oil travel through the bowl to the thickness of the oil layer is 100 to 1 or more, is responsible for the high purifying efficiency realized. The effectiveness and speed of purifying by the centrifugal method as compared with the gravity system permits the continuous purification of circulating oil either by diverting the main stream of oil through the purifier, or by by-passing a portion of it.

The same system of centrifugal purifying of *fuel* oils is employed with a high degree of success. Modifications, however, are required for certain grades of oils, such as a closer spacing of the

separating disks required by the lighter liquids, and the incorporation of a closed system of inlet and outlet connections required in the circulation of liquids containing volatile matter. In the purifying of fuel oil, which usually contains water, and abrasive materials such as sand or silt, the removal of the foreign material in *Diesel* fuel oil reduces the formation of carbon, and reduces the wear and corrosion of the valves, injection nozzles, cylinder liners and feed pumps. In the case of *boiler* fuel oil, the presence of considerable water thoroughly mixed or distributed in the oil would cause no particular trouble, but if passed through in the form of slugs, would lead to serious trouble in the form of re-ignition blasts.

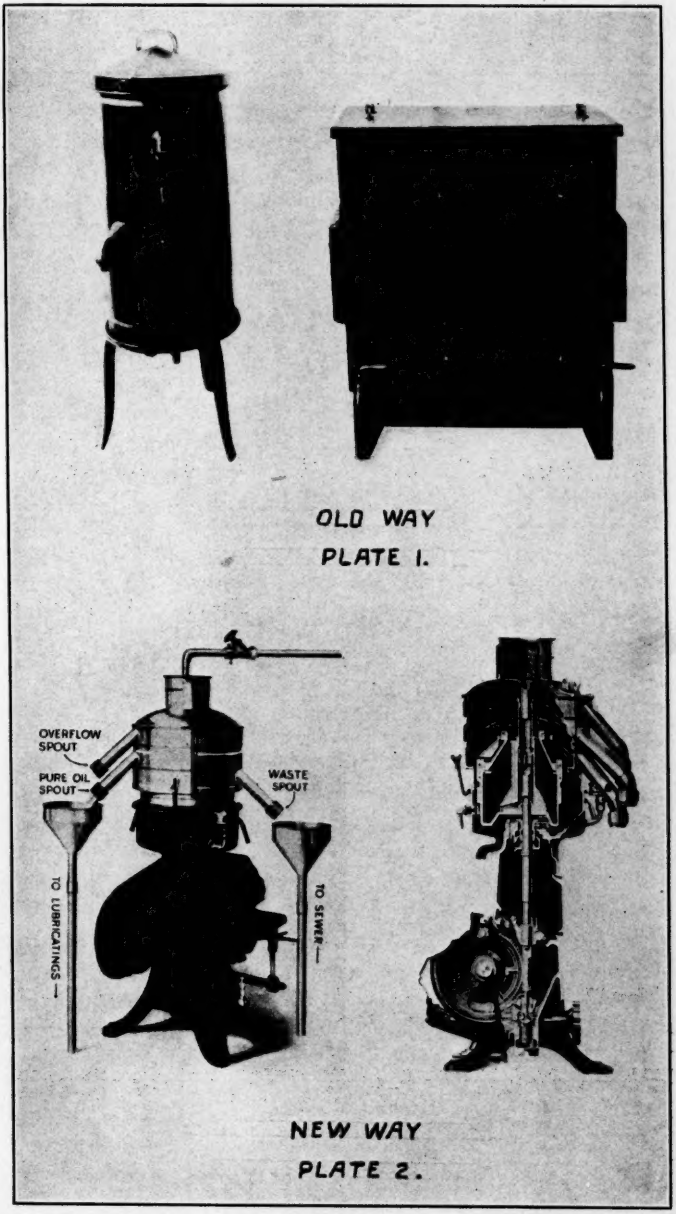
The use of centrifugal purifiers for all types of oil used for lubrication or fuel, especially when stored in bulk, or, as in marine service, where it is stored and used in close proximity to water systems, has proved to be an economic necessity from every standpoint.

LAUNDRY WASHERS.

The subject of laundry washing on Naval ships is one which has existed from the earliest times. Progressive evolution has changed many problems of maintenance, but the problem of laundry washing still continues to be, the expeditious removal of stains, excretions and bacteria.

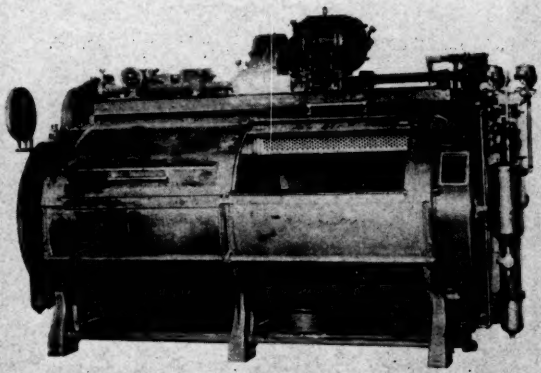
In the former days, washing of clothes was performed by many hands in the usual laborious manner using cold salt water and suitable types of soap. It is evident that the work was necessarily performed on the open decks under the handicap of inclement weather. In the light of modern knowledge of the elimination of bacteria, it is safe to assume that this important item caused no particular concern if the elimination of stains and the appearance of the laundry, measured up to the prevailing standard (see Plate No. 3).

The advent of power driven laundry washer has served not only to practically eliminate the necessity of the large number of individual human washing operations, but with the addition of higher water temperatures, efficient mechanical agitation, and scheduled cyles of operation, a higher standard of sanitary efficiency is made possible within the capacity limits of the machines installed.



Oil Purifiers.

OLD WAY
PLATE 3.



NEW WAY
PLATE 4.

LAUNDRY WASHERS.

The power driven laundry washer used in Naval service on the larger ships, is of the revolving wheel, reversing type (see Plate No. 4). This machine consists of an outer cylindrical shell of rust-proof nickel alloy, riveted and caulked to semi-steel cylinder heads. These heads contain gearing and main support bearings for the inner revolving non-corrodable wheel which is divided into two compartments for the laundry. The revolving wheel is fitted with suitable baffles, perforations in its walls, and sliding doors, so that when filled to its capacity with 350 pounds of laundry, and when filled to the washing level with steam heated fresh water to which washing soaps have been added, the cycle of the washing operation is ready.

The power for revolving the wheel is supplied by a 5 horsepower motor which is started and stopped by the operator, by means of a push button control of the automatic controller. The wheel while operating, revolves at approximately 26 revolutions per minute, and is automatically reversed by the controller every three revolutions. These reversals in direction of rotation serve to greatly increase the efficiency of washing, and prevent the tendency of the materials being washed to become tangled. The machine is equipped with a safety device which prevents operating the motor while the sliding doors are open, and is equipped with an "inching" device to allow the operator to revolve the cylinder to a favorable position so that the sliding doors may be opened for loading or unloading of the laundry. Other equipment such as a water level gage, a thermometer, a fresh water valve, a live steam valve, a soap container tank, and a quick opening drain valve, are supplied to enable the entire operation of the machine to be readily performed by one operator.

The use of machine operation for laundry washing, enables the establishment of a definite cycle of washing and sterilizing in accordance with established laundry practice. The extent to which laundry operations have been perfected with the use of power driven machines, is illustrated by the following cycle of operations as quoted from the manual issued by the Laundry-owners National Association and which has been applied successfully to Navy machines:

1st Bath—Warm water (100 degrees F.); 5 inches of water in wheel after load is saturated. Add the soap and soda mixture to make good suds. Run 10 minutes. Note: The reason why this first bath should not be hot water is, that many food stains and body excretions are albuminous. At a temperature of about 130 degrees to 140 degrees F., the albumin is coagulated, *i.e.*, changed to an insoluble substance.

2nd Bath—Hot water (140 degrees-160 degrees F.); 5 inches of water in wheel. Add the soap and soda mixture to make good suds. Run 10 minutes.

3rd Bath—Hot water (140 degrees-160 degrees F.); 5 inches of water in wheel. Add soap and soda to make good suds. Apply the bleach here. Run 15 minutes.

4th Bath—Hot rinse (140 degrees-160 degrees F.); 10 inches of water. Run 5 minutes.

5th Bath—Hot rinse (140 degrees-160 degrees F.); 10 inches of water. Run 5 minutes.

6th Bath—Hot rinse (140 degrees-160 degrees F.); 10 inches of water. Run 5 minutes.

7th Bath—Warm rinse (120 degrees F.); 10 inches of water. Run 5 minutes.

8th Bath—Cold rinse: 10 inches of water; sour with one-half pint of 56 per cent acetic acid or preferably 6 ounces of sodium acid flouride. Blue lightly. Run 5 minutes.

Draw the load from this bath. Total running time 60 minutes.

The foregoing formula is given for the moderately soiled family bundle (cotton or linen flat-work). For other classes of laundry such as very slightly soiled fabrics, very dirty bundles, white shirts and collars, colored cotton fabrics, overalls, woolens, etc., the formula varies greatly. It is evident that the art of laundry washing has been highly developed by the commercial interests, to produce both quality of work and economy of production. Considerable study of their methods and formulas is necessary by those engaged in laundry work, to adopt the power machines to the limited conditions under which shipboard laundry is required to operate.

The bactericidal value of the power laundry washer is of considerable interest. After all, it is along this line that superiority of the formulated machine driven washing operation is most convincing from the standpoint of the maintenance of the health of large groups of men in close contact, associated with life on board Naval ships. The sample washing formula previously outlined, calls for water temperature of 140 degrees to 160 degrees F. for a collective period of 40 minutes of the one hour total time. It is interesting to note from a report of Dr. G. F. White, re-quoted in part from the manual of the Laundry-owners National Association, the effect of the formula washing temperatures on certain bacteria:—"The approximate thermal death point of representative pathogenic *non-spore-bearing* bacteria is as follows:

Streptococcus—131 degrees F. maintained for 10 minutes.

Micrococcus—140 degrees F. maintained for 10 minutes.

Bacterium diphtheriae—140 degrees F. maintained for 10 minutes.

Bacterium mallei (glanders)—140 degrees F. maintained for 10 minutes.

Bacterium influensae—140 degrees F. maintained for 5 minutes.

Bacterium tuberculosis—140 degrees F. maintained for 20 minutes.

Bacillus typhosus—140 degrees F. maintained for 10 minutes.

Bacillus coli—140 degrees F. maintained for 15 minutes.

Spirillum cholerae—140 degrees F. maintained for 10 minutes.

"Under laboratory conditions, therefore, 140 degrees F. maintained for twenty minutes is sufficient to destroy the foregoing species. What is true for them is true for most, if not all, of the *non-spore-bearing* pathogenic forms. As a working basis, therefore, in laundry problems, a temperature of from 140 degrees F. to 158 degrees F. maintained for twenty minutes would be amply sufficient to destroy all pathogenic *non-spore-bearing* forms. It should be added that the vessel should be a closed one. The temperature and time actually required would depend somewhat upon the care with which the work was done."

From the foregoing, it would appear that the power laundry machine is capable of destroying all *non-spore-bearing* bacteria.

Of the *spore-bearing* bacteria, the report states that "these are destroyed by the action of temperature. Some are killed during the time the temperature is raised to a boil, while others require a boiling temperature for twenty minutes, half an hour or even longer." As it is impracticable to consider operating the power laundry machine at the boiling temperature (212 degrees F.), it is evident that the *spore-bearing* bacteria are eliminated from consideration in the power laundry machine, and recourse must be made to *pressure* sterilizers for protection from spore-bearing bacteria, when the occasion requires this protection.

STEERING CONTROLS.

The subject of the steering of ships dates back ages before machine power, when wind and human strength were used as the agency of propulsion. The first schemes of steering probably were in the use of the side or stern oar methods, followed in later decades by the rudder and helm. Labor saving schemes, such as steam, hydraulic or electrically operated rudder turning machinery, naturally were evolved to keep step with the enormous increase in tonnage of the ship. The *control* of the power machinery which transmits the angular movement of the helmsman's steering wheel to the rudder, is one of the phases of development of the present day. With the increase in efficiency of the many operating devices on modern ships, attention centers on the necessity or the efficiency, of the human control of the steering wheel.

The penalty which must be paid for improper or unsatisfactory steering is in the lengthening of the course and the accompanying retardation of the ship which in turn results in the increase of fuel consumption, wear of machinery, strains of the hull, and maintenance expenses in general. To just what degree the human helmsman can approach the zone of satisfactory steering and the continued maintenance of the same from port to destination, depends on the individual training, morale, sustained observation and application of the mind to the course laid out. The old problem of matching human performance against automatic machine control may still be answered in terms of fatigue, quickness of action, accuracy, amount of movement to restore equilibrium or

OLD WAY
PLATE 5.

NEW WAY
PLATE 6.

STEERING CONTROLS.

the number of re-adjustments, and the extent of overrun of the adjustment.

It is to be conceded that many true and tried helmsmen can give credible hand steering course charts under favorable sea conditions and with proper and sustained application (see Plate 5). Nevertheless, the accomplishment of a properly designed and installed automatic steering control, places the hand steering art under the same comparison as does steam engine governors, automatic voltage regulators, and automatic pressure regulators. Machine controlled devices in general start their functions of control on smaller deviations from the equilibrium adjustment than can possibly be performed by the expected human skill. This assumption is made regardless of the merits of machine supervision for *sustained* periods, compared with human supervision.

The effect of the response of machine control on small deviations results in saving of steering power required of the rudder operating machinery. This is analagous to gyroscopic stabilizers which control the roll of a heavy ship by virtue of the action of a comparatively small gyroscopic mass which operate in anticipation of the normal full roll of the ship. The action of a sensitive steam engine governor likewise performs its function of control of power on small deviations of speed from normal setting. The agency of human control can hope to compete with machine control, only where conditions of operation are so varied and unusual that no definite point of equilibrium is presentable for any appreciable length of time; as for example in certain maneuvers of a ship, the control of hoisting engines, or the steering of vehicles in traffic.

The modern machine control of steering is exemplified in the automatic gyro-pilot which is being used in increasing numbers on merchant vessels and Naval ships. The elements of this device, shown in Plate 6, consist of a roller contact maker operated by a repeater motor electrically connected to the Master-Gyro-Compass; a follow-up segment ring which is operated as the rudder moves; and a power device controlled by the contact-maker, which operates the ship's wheel or steering engine valve. The elements of this simple mechanism effect a follow-up control of the rudder in which the angle of applied rudder is proportional to the ship's angular departure from the course.

When steering automatically, with the ship on its course and the rudder amidship, the roller contact maker rests on the insulated segment of the follow-up ring. When the ship's head falls off the course, the roller contact is moved off the insulated segment by the corresponding movement of the gyro repeater motor, completing a control circuit through the follow-up ring which closes an electromagnetic relay. This, in turn, starts the steering motor and results in a movement of the rudder to bring the ship back to the course.

The steering motor is electrically or mechanically geared to the follow-up mechanism so that when the motor turns, its movement is transmitted to the follow-up ring, causing it to follow or "chase" the roller contact maker, and thus bringing the insulated segment over the roller. This opens the control circuit, the electromagnetic relay falls out, opening the steering motor circuit and closing a dynamic brake contactor which prevents the steering motor from over-running. When the ship's head falls off in the other direction, the roller contact maker is moved to the opposite side, energizing a circuit that reverses the steering motor and causes the rudder to move in the opposite direction.

A small pilot wheel is provided to permit movement of the roller contact maker when it is desired to make alterations in the course during automatic steering. For vessels which must maneuver in either direction with speed and accuracy, a type of control is provided whereby course changes up to 180 degrees may be instantly set in with the pilot wheel.

In actual practice it is necessary to provide for several factors which have not been mentioned in the elementary principles outlined above. For instance, when rudder is applied, it is best to make the first application large enough to quickly check the tendency to leave the course. When the ship returns to her course, it is necessary to "meet her" with a rudder movement which will check the swing and steady her on the course. Some ships are more sensitive to rudder movements than others and provision must therefore be made for adjusting the ratio of rudder movement of ship's head. When operating in a seaway, it is desirable to allow the vessel to yaw through a small angle without bringing the steering gear into operation. This eases the steering engine and prevents the vessel from swinging with a rapid, forced yaw.

For these reasons, adjusting devices are provided on the Gyro-Pilot control unit, which introduce a greater or lesser amount of lost motion between the follow-up mechanism and the follow-up ring, and between the gyro repeater motor and the roller contact maker. By a simple manipulation of the adjusting knobs, the Gyro-Pilot may be set for the roughest condition of wind and sea, and more accurate courses, with less wear and tear on the steering machinery, can be steered automatic than could possibly be attained by the average human helmsman.

In every installation of the Gyro-Pilot, its rudder control is arranged to be instantly and positively connected or disconnected at the ship control station, leaving the ship's gear without alteration or hazard, for emergency use. The choice between ship's gear control, Gyro-Pilot manual control, or Gyro-Pilot Automatic Control, is thus instantly available.

VOICE PROJECTORS.

The present day methods for holding long distance conversation by means of the wired telephone or by radio telephone, have been highly successful as a means of direct communication. The practical limitations of the wired telephone is in the necessity of establishing the wire connection between the sending and the receiving ends. With the radio telephone, the practical limitation is the amount and complexity of the apparatus required, together with the lack of flexibility and directiveness for *short* distances.

One of the earlier means of extending short range conversations was by the use of the cupped hands. This was followed in recent time by the use of hand megaphone. The latter device still serves many purposes such as hailing near-by boats, docking, mooring, transmitting orders on deck, and contact work in general where the distance is not more than a fractional part of a mile, and where the extraneous noises are of a low order of intensity (see Plate No. 7).

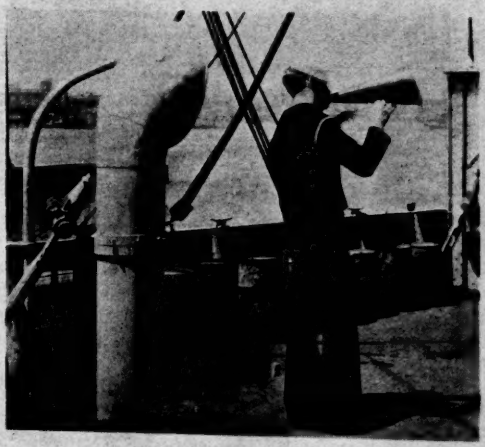
New conditions have arisen in Naval Service which require direct communication beyond the scope of the hand megaphone. This is with reference to the desirability for establishing direct communication with the pilots of aeroplanes in mass formation

on the decks of aeroplane carriers prior to flight; for establishing communication between dirigibles and ground crews; for hailing boats or aeroplanes; and other similar application of intermediate distances beyond the range of direct voice calling.

For the intermediate range of communication there has been recently evolved a high intensity sound projector (see Plate No. 8), which functions as follows: The human voice is picked up by the usual electrical microphone receiver where the battery powered voice currents are generated. These currents are electrically amplified and then are wired to a receiver attached to a large acoustic projection horn. The projector is equipped with an air chamber to which is supplied compressed air at 25 pound pressure per square inch, furnished by a special compressor or from the plant source. The compressed air discharges through an orifice in throat of the projector in varying amounts depending on the movement of an electromagnet valve which is operated by the magnetic pull produced by the voice currents.

The mechanical operation of the voice valve and projector is analagous to the mechanism of the human voice where the moving air from the lungs is modulated by the muscular control of the larynx to produce resonant sound in the cavities of the head which in turn are projected through the open mouth. The effect of the electric voice impulses on the air discharge into the throat of the projector is a high degree of sound amplification, the power of which is sufficient to project ordinary speaking voice, at a horizontal distance of 3000 feet under usual wind conditions. The projected sound has a fairly narrow beam, and when directed at an airship at 3000 feet elevation, an intelligible transmission has been obtained for a horizontal distance of nine miles from the source.

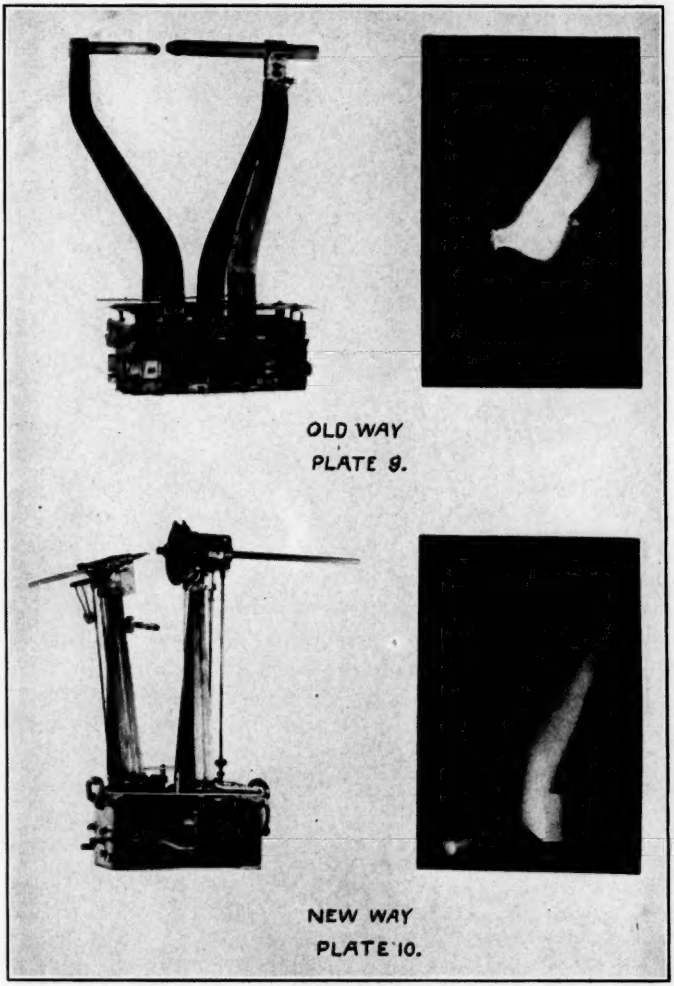
By the use of this form of intense amplification projected through one or as many as eight valves in parallel, it is possible to transmit direct voice communication from ship to ship equipped with these devices, or for one-way communication from ship to small boats, landing decks, aeroplanes in flight, etc. For secrecy of communication, the direct voice projection, within its range, has advantages over radio communication for the reason that there



OLD WAY
PLATE 7.

NEW WAY
PLATE 8.

VOICE PROJECTORS.



SEARCHLIGHTS.

is a reasonably definite range for the sound waves of voice projection, whereas the radio system can not definitely be limited as to range of transmission due to the fact that the faintest words or signals, which are in the form of electric waves, may be picked up and amplified at very distant points from the source.

SEARCHLIGHTS.

Searchlights in the Navy have been used for many purposes including signaling, as an aid to navigation and for target illumination. Prior to the World War, these lights were used only with relatively short range guns, at ranges not exceeding 3000 yards, principally for defense against destroyers. During the World War, torpedo ranges were increased, and aircraft were added to the list of offensive armaments. It therefore became necessary to illuminate targets in the air as well as on the surface of the sea.

The pre-war lights were what are classed at the present time as *low* intensity lights (see Plate No. 9). This classification was based on comparative intensity of the type of arc used. The arc was of a type wherein practically all of the illumination emanated from the incandescent carbon in the crater of the positive electrode, and the incandescent carbon tip. Placing the crater in the focal center of a parabolic mirror produced a beam of light in the reddish-yellow spectrum. Naturally the temperature at which the carbon volatized was the limit of intensity. No matter how much additional power was used, no increase of intensity was accomplished above the critical point. The low intensity lights in sizes of 12 inches, 18 inches, 24 inches, 30 inches, 36 inches, 48 inches and 60 inches were carried to their ultimate usefulness in development, but eventually fell short of modern requirements. The optical system consisting of a silvered glass mirror and a flat glass front door, with an efficiency of more than 90 per cent, permitted very little advance along this line.

In 1916 the development of the light was attacked from a new angle. A new type positive carbon electrode was presented, in which the current density of the arc could be greatly increased over the old type, resulting in a phenomenal increase in beam

intensity (see Plate No. 10). The new electrode consists of a hard baked shell of carbon made from anthracite coke, surrounding a core made of a mixture of bituminous coke and cerium fluoride. The higher electrical conductivity of the core material, compared with the shell, results in the concentration of the arc current of high density per unit area of the crater. This in turn generates an intense, incandescent, superheated, current conducting gas-ball, by virtue of the volatization of the cerium fluoride. The gas-ball is held in the confines of the crater by the force of the electron stream emitted by the negative electrode. The negative electrode is mechanically placed at an angle of 15 degrees to the positive electrode, to produce this phenomenon. This high intensity arc also produces a beam color far more adaptable to the service, transferring the beam from reddish-yellow to the efficient blue-green spectrum, the latter appearing white to the eye.

In the old 36 inch low intensity lights, the electrodes are 1¼ inches diameter for the positive and 1 inch diameter for the negative, with a resultant positive crater diameter of approximately ⅞ inch. The maximum usable power is 120 amperes at 60 volts at the arc or 7200 watts. This produces about a 40,000 arc candle power. In the new 36 inch high intensity lights, the electrodes are .630 inch diameter for the positive and .433 inch diameter for the negative with a resultant crater diameter of approximately ½ inch. The maximum usable power is 150 amperes at 80 volts at the arc, or 12,000 watts. This produces about 140,000 arc candle power. Thus with a 65 per cent increase in power a 250 per cent increase in light intensity is realized.

In the old style lamp, the arc regulating mechanism comprises two electrode holders mounted on separate carriages, moved horizontally apart or together, to maintain a predetermined voltage across the arc. The mechanism functions from abnormal arc voltage on a magnetically driven double ratchet, operating feed screws engaging the carbon holder carriages. A predetermined rate of feed of positive and negative in the geared ratio of 3 positive and 2 negative, is maintained, and irregularities of carbon consumption different from this ratio are compensated for, by hand adjustment of the lamp, to bring the positive crater into the mir-

ror focus. With the development of the new arc, much more critical adjustment to maintain the gas-ball, became necessary. In this arc, the positive electrode is rotated constantly in order to keep the crater shape uniform. The positive is automatically fed regardless of its consumption, controlled by a thermostat to keep the positive tip, always in a predetermined position outside the positive head. The negative electrode is fed in and out of the arc, and is controlled by motor-operated clutches, which are engaged as needed by a solenoid voltage regulator. This negative electrode and its mechanism maintains the predetermined arc voltage very accurately, so that optimum power is obtained at the arc.

The accuracy of arc regulation, setting of the electrodes relative to each other, the high current density at which the electrodes are operated, the positive electrode core composition, and gas-ball operating at temperatures far above the volatizing temperatures of carbon, have resulted in the superseding of the old arc whose light depended upon hot carbon. Thus through a radical change in principle, a new high intensity arc has been developed, which has brought the searchlight on a plane of efficiency with an effective range of 8000 yards under good atmospheric conditions. The old arc, on account of the tendency of the crater to wander, had a very limited elevation during operation. The new arc, on account of the force of the electron stream, will function successfully in any position of elevation up to vertical, and is therefore well adapted to aircraft defense.

All of the progress in searchlight development has been with respect to the arc. The optical system has not been changed at any point in the progression except to reduce the breakage of the glass used in the mirror and the front door. In the former item, polished metallic mirrors are being considered, while as a substitute for the flat glass strips formerly used in the front door, a special form of a one-piece dome-shaped glass, is now the accepted development.

SHIP TRIAL ANALYSIS TO OBTAIN PROPELLER DESIGN FACTORS.

By Commander H. F. D. Davis, U. S. Navy, Member.

INTRODUCTION.

1. The fact that ship hull and ship propeller design do not yet rest on completed foundations grounded in theoretical hydrodynamics has been pointed out by many authorities. Present day designers in these fields therefore generally rely on the information obtainable by experiments with models of both hulls and propellers, each alone and also the two combined in self-propelled tests. Model methods of investigation in these fields were developed largely by William Froude and his son, R. E. Froude, both of whom stand at the very head of the list of distinguished pioneers in engineering of the last century of progress. It should be noted that, in accordance with the "law of dynamical similarity," first enunciated by Newton, a thoroughly scientific attack is made possible by means of models on these problems as on many others which bother engineers. Mr. R. E. Froude explained methods and gave the results of model tests in papers the most important of which were read by him before the Institute of Naval Architects in 1883, 1886 and 1908. Model methods quite naturally came into use because they are relatively inexpensive as compared to full size experiments of the same scope.

2. Another source of information for designers lies in the records of performances of actual ships. Because of the relatively enormous expense of such work as compared with the cost of similar work with models, relatively little "systematic" investigation is possible with full scale ships and propellers. However, the results obtained in the full scale trials that are held serve to check at many points the results obtained in "systematic" model tests, and to give designers measures of the amounts by which model results must be modified to accord with actual practice.

3. Model tests and ship trials may thus be made complementary to one another and it is believed that to be most useful they should be developed and prosecuted in closely parallel and directly comparable lines. Another desideratum in a system of propeller design based on experience with ships and models as herein discussed, is that it should be two-way: analysis one way for the design factors, design the other way using those factors.

4. It is the purpose of this paper first to review the Froude methods, particularly for those who do not have access to the original papers, with a view to showing how they serve in the investigation of ship propulsion problems, both of the model and of the prototype, the treatment being purposely parallel for the model and the ship. Having developed a basis by means of this review, formulations for analyses of ship trials are presented through examples, showing how design factors may be obtained from ship trials. While the design features of the Froude system have been emphasized in many articles on the subject, herein analysis is specially stressed.

BASIC RELATIONS IN THE PROPULSION PROBLEM.

5. Mr. R. E. Froude gives[1] the following mathematical relations and definitions for detailed investigations of the propulsion problem of the ship or its model: (The following applies to a single screw ship or one shaft of a multiple screw ship.)

At a speed V, of a ship or model

$$\frac{E}{P}\ \frac{\text{(Effective horsepower)}}{\text{(Shaft horsepower)}} = \text{"propulsive coefficient"} \qquad (1)$$

$$\frac{U}{P}\ \frac{\text{(Useful horsepower)}}{\text{(Shaft horsepower)}} = \text{"propeller efficiency"} \qquad (2)$$

$$\frac{E}{U}\ \frac{\text{(Effective horsepower)}}{\text{(Useful horsepower)}} = \text{"hull efficiency"} \qquad (3)$$

$$\text{from which } \frac{E}{P} = \frac{E}{U} \times \frac{U}{P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

E^2 = the effective horsepower; the power required (if developed

[1] Trans. I. N. A. 1883. Trans. I. N. A. 1886.

[2] The notation herein generally follows that used by Rear Admiral D. W. Taylor (C. C.) U. S. N. Retired, in his recent papers, *e. g.*, *The Shipbuilder*, April, 1930.

through a tow rope for example, no propellers being installed) to maintain the speed V of ship or model.

U = (propellers being in use) the thrust or useful horsepower developed by the propellers in maintaining speed V of ship or model or the thrust horsepower developed by an "open" propeller at speed of advance V_A defined below as being related to speed of ship V in a particular way.

P = the shaft horsepower required in (absorbed by) the propeller shafts in maintaining speed V of ship or model; P being measured at the torsionmeter, frictional losses between torsionmeter and propeller, usually small and unknown, being neglected. If indicated horsepower (I.H.P.) is measured, P must be obtained by applying a measured or estimated value of mechanical efficiency, e_m. Thus $P = \text{I.H.P.} \times e_m$.

6. The above mentioned power units may each be expressed in more fundamental English system units as follows:

$$E = RV \times \frac{6080}{60} \times \frac{1}{33000} = RV \div 326 \quad . \quad . \quad . \quad . \quad (5)$$

$$U = TV_A \div 326 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$P = 2\pi NQ \div 33000 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

in which

R = pull in pounds in the tow rope; also the resistance of model or ship in pounds, inclusive of all appendage resistance.

V = speed in knots per hour of model or ship relative to still water.

N = revolutions per minute of propeller shaft and propeller.

Q = torque in foot pounds in the propeller shaft.

T = thrust in pounds developed by the propeller.

V_A = the "mean uniform speed of advance" of the propeller through the wake water in the case of self propelled models or ships—a mathematical convention; and through undisturbed water in the case of so-called "open" model propeller tests.

7. Extensive model tests have established the following relations:

$$V_A = V(1-w); \quad \frac{V_A}{V} = 1 - w; \quad w = 1 - \frac{V_A}{V} \quad . \quad . \quad (8)$$

$$R = T(1-t);\ \frac{R}{T} = 1-t;\ t = 1 - \frac{R}{T} \quad . \ . \ . \ . \quad (9)$$

w being defined as wake [3] fraction and t being the thrust deduction fraction, two factors which are necessary to account for the interaction of hull and propeller.

8. The hull efficiency, substituting (5) and (6) in (3), then using (8) and (9),

$$\frac{E}{U} = \frac{RV/326}{TV_A/326} = \frac{R}{T} \times \frac{V}{V_A} = \frac{(1-t)}{(1-w)} \quad . \ . \ . \ . \quad (10)$$

9. These ten equations with six or seven more to be given later contain the whole meat of the ship propulsion problem.

THE APPLICATION OF THE FOREGOING RELATIONS TO MODEL EXPERIMENTS.

THE DETERMINATION OF EFFECTIVE HORSEPOWER, E.

10. If a *ship model without propellers* is towed in a model basin, such as that at the Navy Yard, Washington, D. C., the resistances R_1, R_2, etc., corresponding to speeds V_1, V_2, etc., can be easily measured, and the effective horsepowers may then be calculated with a very small error, using formula

$$E = RV \div 326 \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad (5)$$

A curve of effective horsepower, E, vs. speed V, can then be plotted in the usual way.

11. Carefully conducted experiments and investigations have shown that the total resistance of a model at speed V can be broken down, for a purpose made clear later, into two components: (a) the frictional resistance which can be computed from the relation

$$R_f = fSV^{1.83} \quad . \ . \ . \ . \ . \ . \ . \ . \quad (11)$$

in which f is a friction coefficient experimentally determined and S is the wetted surface in square feet; and (b) the residual resistance R_r which is made up of the wave making, eddy and air resistances. With well streamlined designs the eddy and, in the

[3] This definition of wake fraction is that of Admiral Taylor, not the one used by Froude.

model basin conditions, the air resistances comprise a very small part of the total.

12. The total resistance R of the model having been measured and the frictional resistance R_f computed, the residual resistance of the model is obtained from the relation

$$R_r = R - R_f \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

MODEL PROPELLER "OPEN" TESTS.

13. A model propeller may be set up for "open" tests and torque, Q, thrust, T, speed of advance, V_A, and revolutions per minute, N, may be measured throughout the useful working range, from which data coordinates of the so-called propeller characteristic curves may be computed and plotted.

14. The coordinates of the propeller characteristic curves may be computed and plotted in many different ways. The form used by the Model Basin in Washington for single propellers shows torque coefficient, thrust coefficient and efficiency plotted on true slip, the coordinates used for plotting being obtained from the test data with the following formulae:

$$C_Q = \frac{Q}{n^2 p^3 d^2}, \text{ torque coefficient} \quad . \quad . \quad . \quad . \quad (13)$$

$$C_T = \frac{T}{n^2 p^2 d^2}, \text{ thrust coefficient} \quad . \quad . \quad . \quad . \quad (14)$$

$$e = \frac{T_v}{2 \pi Q n}, \text{ propeller efficiency} \quad . \quad . \quad . \quad (15)$$

$$s_t = \frac{pn - v}{pn}, \text{ true slip} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

wherein

$$n = \frac{N}{60} = \text{revolutions per second.}$$

$$v = V_A \times \frac{6080}{3600} \text{ speed of advance in feet per second.}$$

d = diameter of propeller in feet.

p = face pitch of propeller in feet.

Since C_Q, C_T, e, and s are non-dimensional, a set of such curves for a model propeller can be applied to any similar propeller assuming the "law of dynamical similarity" to apply. In other words, for a propeller of any diameter, if its pitch ratio, $p \div d$, its helicoidal area ratio [4] (or projected area ratio or mean width ratio) is the same as that of a tested model, the torque, thrust and efficiency when it is working at any given true slip, may be computed using these formulae after selecting the appropriate coefficients from the curves for such model.

15. Another set of non-dimensional equations used by Admiral Taylor in some recently published work [5] on propeller characteristic curves is as follows:

$$B_P = NP^{\frac{1}{2}} \div V_A^{5/2} \text{ (basic S.H.P. variable)} \quad . \quad (17)$$

$$B_U = NU^{\frac{1}{2}} \div V_A^{5/2} \text{ (basic U.H.P. variable)} \quad . \quad (18)$$

$$e = U \div P \text{ (propeller efficiency)} \quad . \quad . \quad . \quad . \quad (15)$$

$$\delta = Nd \div V_A \text{ (speed coefficient)} \quad . \quad . \quad . \quad . \quad (19)$$

Using these formulae and the data from model basin experiments on a series of 3-bladed propellers, pitch ratios 0.5 to 1.8, of mean with ratio (MWR) .25 (equivalent helicoidal area ratio, $\frac{HA}{DA}$ = .38 and projected area ratios, $\frac{PA}{DA}$ = .29 to .35) and with blade thickness fraction .05, characteristic curves have been worked up and plotted on two sheets. Similar sheets are also available for

[4] The helicoidal (or developed) area ratio, HA ÷ DA, of a propeller is the ratio of the developed area of the blades outside the hub to the area of the disc swept out by the propeller, DA = $\pi d^2 \div 4$. The projected area ratio, PA ÷ DA, is similarly the ratio of the projected area of the propeller blades (outside the hub and projected on a plane perpendicular to the axis) to the disc area. The mean width ratio, h, is defined by the relation developed area outside the hub = mhd^2, md being the length of blade outside the hub. It may easily be shown that HA ÷ DA = 1.274 n h m, n being the number of blades; m varies from 0.833 for solid propellers to 0.80 or even less for built-up propellers.

[5](a) "Propeller Design Developments"—Rear Admiral D. W. Taylor (C. C.), U. S. N. Retired—World Engineering Congress, Tokyo, 1929.

(b) "Model Results of a Four Bladed Propeller Series"—Rear Admiral D. W. Taylor (C. C.), U. S. N. Retired—American Society of Naval Architects and Marine Engineers, 1929.

4-bladed propellers.[6] These curves are in a form particularly useful to the propeller designer because the formulae deal directly in horsepower, speed of advance in knots, and R.P.M., *i.e.*, the data as obtained from standardization or other trials. These curves are plotted in somewhat more convenient form than the equivalent "delta rho" diagrams found in the Transactions of the Society of Naval Architects and Marine Engineers for 1924 and in Admiral Taylor's "Speed and Power of Ships."

SELF-PROPELLED MODEL TESTS.

16. Assuming that a hull model has been towed and the model propeller has been tested as described, suppose we combine the hull and propeller models in self propelled tests; revolutions per minute, N, torque Q, speed V, and thrust T may be measured very accurately with the experimental apparatus now developed, from which data the shaft horsepower P may be computed directly using formula

$$P = 2\pi NQ \div 33000 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and thrust deduction t using formula

$$\frac{R}{T} = 1 - t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

But, since neither V_A nor w can be directly measured, a roundabout way must be taken to determine them and the other results needed for a complete analysis.

17. Froude's method of determining the wake with models involves

(a) The testing of the model propeller "open," *i.e.*, without the interference of the hull, measurements of N, Q, T, and V_A being taken to obtain the "open" characteristics of the model propeller as described in paragraphs 13 to 15.

(b) The testing of self-propelled model as described in paragraph 16 to measure N, Q, T and V.

[6] "Model Results of a Four-bladed Propeller Series"—Rear Admiral D. W. Taylor (C. C.), U. S. N., Retired—Transactions of Society of Naval Architects and Marine Engineers, 1929, Plates 34-42.

18. It is then assumed, the assumption being justified by much experimental work,[7] that, for identical values of N, Q, and T obtained both in the self-propelled model and "open" model propeller tests the corresponding values of V and V_A correspond to each other and the wake fraction may accordingly be obtained from the relation

$$(1 - w) = V_A \div V \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

19. For emphasis it may again [see formula (16)] be noted that in the "open" propeller and self propelled model tests the measured face pitch of the model propeller p, V_A and N are connected by the relation

$$V_A \times \frac{6080}{60} = N \times p \times (1 - s_t) \quad . \quad . \quad . \quad (16)$$

V_A being the measured speed of advance of the propeller through undisturbed water of course. s_t is the *true* slip (fraction).

20. The above relation should be compared with the expression connecting the measured face pitch of a propeller p, the speed of the self-propelled model V, and the *apparent* slip (fraction) s_a:

$$V \times \frac{6080}{60} = N \times p \times (1 - s_a) \quad . \quad . \quad . \quad (20)$$

21. It is evident from the foregoing that the application to self-propelled ship models of the Froude method of investigation is

[7] With respect to this assumption, which does not appear to have been seriously questioned, Froude states in his 1883 paper:

"What are the facts yielded by the experiments behind model and in still water, taken together? We have (1) the thrust and revolutions of the screw working speed V behind model; (2) the thrust and revolutions, each identical with the foregoing, at speed V_1 in still water; (3) the turning moment at the same speed in still water, which we generally state in the form of turning force as measured for efficiency. We then infer, on the assumption of a uniform wake, that the turning moment behind the model will be the same as in still water, and that the turning force will therefore be less in the ratio in which V_1 is less than V. But we also have, recorded by the experiment (4), the actual turning moment behind model, which in the same manner we convert into turning force; and the degree of agreement between this actual turning moment or turning force (it does not matter in which form the comparison is made) with that calculated from the experiments in still water, is a criterion of the sufficiency of the assumption that the disturbance of water which we term the "wake" amounts in effect to a mere uniform forward current. Now this comparison has been made in a vast number of experiments, including a great variety of conditions differing in character and amount of wake, and the agreement, though not in all cases quite exact, is so nearly exact as almost to be covered by the limits of error of the observations. The general direction of this difference, such as it is, is the opposite of what one would be at first inclined to expect, since it shows that the actual turbulent wake left by the model increases the efficiency of the screw more than would be hypothetical uniform wake, the efficiency given by the turning force recorded in the experiment behind the model being higher than that calculated from the experiments in still water by an amount varying from *nil* to about 2 per cent, in all the various conditions which have been subjected to experiment. I believe the variations, at any rate, in the amount of this difference, to be due chiefly to inaccuracies in the record."

exceedingly direct and relatively simple, involving only one assumption in order to determine the values of speed of advance V_A and wake fraction w which are necessary in order to connect the behavior of the propeller "open" with that "behind"; this assumption being in effect that there is a uniform value of water flow V_A which is equivalent in effect to the confused flow which arises behind a model.

22. The complete analysis of the model performance can now be completed:

R and V having been measured as described for the hull model

$$E = RV \div 326 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

T, N, Q and V_A being obtained from the model propeller tests and T, N, Q and V having been measured in the self-propelled model tests $(1 - w)$ is determined by

$$V_A = V\,(1-w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$\text{also } \frac{R}{T} = 1 - t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$P = 2\pi NQ \div 33000 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$\text{and } U = TV_A \div 326 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Then "propulsive coefficient" $= \frac{E}{P}$, "hull efficiency" $= \frac{E}{U}$, "propeller efficiency" $= \frac{U}{P}$; thus all the essentials of the problem are determined.

THE APPLICATION OF THE FROUDE METHOD TO FULL SIZED SHIP TRIALS.

23. It has been shown that the Froude method of investigation can be applied directly to model tests by making a single assumption with respect to the use of the model propeller characteristic curves which assumption, however, has been very fully justified experimentally. Can the method be applied to ship trials? What are the limits of such application? The discussion will follow that given above for models.

THE DETERMINATION OF EFFECTIVE HORSEPOWER, E.

24. Short consideration of the problem is required to make clear that the towing of full-sized ships to determine the resistances R, R_1, R_2—at the corresponding speeds V, V_1, V_2—is impracticable. In meeting this situation, however, William Froude,[8] the father of R. E. Froude, under authority of the British Admiralty, directed the towing of H.M.S. *Greyhound* by H.M.S. *Active* with complete arrangements for measuring the tow rope pull and the speed. The *Greyhound* was a "screw sloop" 172.5 feet on the water line, displacing 938, 1050, and 1161 tons during different trials which extended through six weeks. The speeds were as high as 13 knots.

25. The results of these trials when compared with tests in the Model Basin of a 1/16 scale model of the *Greyhound* established on an experimental basis, the Froude "Law of Comparison."[9]

26. Following the epoch making tests of the *Greyhound*, in the studies of the results it was shown: (a) that the frictional resistance of a ship does not follow the above mentioned law but may be computed from the formula (11).

$$R_f = fSV^{1.83} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

(Other formulas have been suggested by subsequent studies of the resistance problem.)

(b) that the wave making resistance does follow the law;

(c) that with good design and under ideal trial conditions the eddy and air resistances form such a small part of the residual resistance that the total resistance of the ship may be divided into two parts as in the case of the model, and, as a result of the above mentioned conditions, that the total resistance at speed V of a full sized ship may be arrived at by

First, computing the frictional resistance of the ship by using $R_f = fSV^{1.83}$ or other formula;

[8] Transactions I. N. S. 1874.

[9] The "Law of Comparison" developed and applied by Froude to ship model tests is a special application of the "law of dynamical similarity" first stated by Newton and later developed by other mathematicians. The Froude law may be stated as follows: for bodies in motion "mechanically and geometrically similar, if the speed ratio is made equal to the square root of the dimension ratio, all external forces must be in the ratio of the cube of the dimension ratio." (Speed and Power of Ships, page 28). See also "Model Experiments and the Forms of Empirical Equations"—E. Buckingham, Transactions of American Society of Mechanical Engineers, 1915.

Second, computing the residual resistance of the model as described;

Third, computing the residual resistance R_r of the ship from the residual resistance of the model by applying the Froude "Law of Comparison";

Fourth, adding these ship frictional and ship residual resistances to make up the total

$$R = R_f + R_r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Then

$$E = RV \div 326 \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$= \text{the effective horsepower of the ship at speed V}$$

27. The method described above has been criticized from many angles, entirely different procedures, different constants, different ways of taking into account the appendage resistance, etc., having been suggested by the different investigators and students of the problem. For the present purpose, however, the foregoing may suffice to give ideas of the difficulties involved and the assumptions which must be made in arriving at the probable resistance (effective horsepower) of a full sized ship at any given speed. It must therefore be emphasized that, due to the assumptions made in this process, the resistance and the effective horsepower so arrived at are only applicable when the design is good and trial conditions are ideal. For other conditions it may be practicable, when enough data and experience have been collected, to estimate a proper effective horsepower by simply adding to the figure for ideal conditions percentages to allow for rough water, wind, and foul bottom. Few published data of this kind are, however, now available.

FULL SIZE PROPELLERS.

28. In the same way that full scale tests of model hulls are impracticable, "open" water tests of full size propellers are out of the question because of the very large forces and volumes of water involved. Since the shaft horsepower absorbed, the useful (thrust) horsepower developed and the efficiency of full size propellers are required in a solution of the propulsion problem, such information must be obtained in the indirect way described before in giving the explanation of model propeller experiments.

THE APPLICATION TO SELF PROPELLED SHIP TRIALS.

29. As in self propelled model tests, so during ship trials the revolutions per minute, N, the torque, Q, the speed, V, and the thrust, T, may be measured with reasonable accuracy, using instruments now available. In carefully conducted trials the revolutions per minute, N, the torque, Q, and the speed, V, have been measured; but very unfortunately, thrust measurements have not often been made, although essential, as was seen above for the model, to the best solution of the propulsion problem. With the measurements mentioned above, including thrust T, computations similar to those indicated for the self-propelled model may be made, viz:

$$P = 2\pi NQ \div 33000 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

$$\text{and } \frac{R}{T} = 1 - t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

30. Still paralleling the discussion for models, it is found that no direct method is available for measuring the speed of advance V_A or wake fraction w during ship trials; "open" tests of full sized propellers also being impracticable, it is not possible to follow the plan used in the case of the model and, if the problem is to be solved, another assumption must be made. Again, it is the 'Law of dynamical similarity," that is called upon, and by virtue of this law the assumption is made that the characteristic curves obtained from "open" model propeller tests may be used for the full-sized propellers. Then, as described before for the models, for identical values of N, Q, T obtained in the ship trials and taken from the characteristic propeller curves, the measured V of the ship trials and the corresponding V_A from the characteristic curves, correspond to each other and are related by the equation

$$V_A = V\,(1 - w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

31. In taking up further discussion of analyses of ship trials the subscript m will be used to distinguish values actually measured during the trials from those taken from the characteristic curves, *e.g.* Q_m (P_m), T_m (U_m) vs Q (P), T (U).

32. In the analysis of ship trials it is evident that three different cases may arise:

(a) Case I—Both the torque (P_m) and thrust (T_m) may be measured, the "given" data in this case comprising

1. Propeller dimensions and characteristics
2. Speed, V
3. Revolutions, N
4. Thrust, T_m
5. Shaft horsepower, P_m (Q_m)
6. Effective horsepower, E; the effective horsepower obtained as described in paragraph 26 or by other estimate is also included as "given" data.

(b) Case II—Same as (a) except thrust, T_m, is not observed, this case being the most usual. Not having T_m the approximation value T, taken from the propeller characteristic curves and corresponding to P_m (Q), must be used for a complete, albeit approximate, solution.

(c) Case III—Same as (a), except shaft horsepower, P_m, is not observed, this being a possible case. As in Case II so in this case, P_m (Q) not being available, the approximation P (Q), taken from the propeller characteristic curves and corresponding to T_m, must be used for the complete, albeit approximate, solution.

In Case I which parallels the self-propelled model tests very closely, but unfortunately is not common, the characteristic curves are required only for the purpose of determining V_A and the wake fraction w, all the other relations being determined by the observed data. Thus $E \div P_m$ and $t = 1 - (R \div T_m)$ come directly from the trial data; V_A is next obtained by use of the propeller characteristic curves, and since $U_m = T_m V_A \div 326$, hull efficiency ($E \div U_m$) and propeller efficiency $U_m \div P_m$ then follow immediately.

33. In Case II the observed data determine:

Propulsive coefficient $E \div P_m$ directly and with the assistance of the curves first V_A and w as in Case I; then T(U), $E \div U$; $E\ (1 - w) \div U$ and $U \div P_m$ these values being approximations involving a relation between the propeller efficiency "open" and "behind" which has been called the "relative rotative efficiency" equaling

$$\frac{U_m}{U} = \frac{T_m}{T}; \frac{P}{P_m}$$

34. In Case III the observed data determine directly only the thrust deduction fraction $t = 1 - (R \div T_m)$ then with V_A and w the hull efficiency $E \div U_m$ and with further use of the curves

$$U_m \div P \text{ and}$$
$$E \div P$$

both of which relations also involve the "relative rotative efficiency"

35. In order to summarize and make more clear the foregoing, the following tabulation is given:

(a) Results which come from observed data with no use of the curves:

Case I	*Case II*	*Case III*
$E \div P_m$	$E \div P_m$	
$R \div T_m = (1-t)$		$R \div T_m = (1-t)$

(b) Results following use of curves to determine V_A and $(1-w)$ "relative rotative efficiency" not involved:

V_A (w)	V_A (w)	V_A (w)
$U_m = T_m V_A \div 326$		$U_m = T_m V_A \div 326$
$E \div U_m$		$E \div U_m$
$U_m \div P_m$		

(c) Approximations to the foregoing, with further use of curves, "relative rotative efficiency" being involved:

U (approx.)	
$E \div U$ (approx.)	
$U \div P_m$ (approx.)	
	P (approx.)
	$U_m \div P$ (approx.)

36. The great advantage of making both the torque and thrust measurements is well shown by the foregoing comparisons and cannot be too greatly emphasized. As previously shown, in this

case the propeller characteristic curves are required only for the purpose of determining the wake fraction. The wake fraction having been determined, all other results follow directly from the measured data and only contain such errors as may exist in these data. More important even, the torque and thrust measurements serve to check one another.

37. Although trials as they have been usually conducted give less data than are desirable, the results when only torque is measured may be analysed as shown above for Case II. Poor as these data may be relatively, they are the best available from most trials and must therefore be utilized if design factors from such ship trials are desired.

38. It may properly be pointed out, emphasizing what was said before, that the effective horsepower, E, enters into the numerators of the following expressions:

	Propulsive coefficient	$E \div P_m$
Approximation to	Propulsive coefficient	$E \div P$
	Hull efficiency	$E \div U_m$
Approximation to	Hull efficiency	$E \div U$
	1—t	$R \div T$
Approximation to	1—t	$E\ (1-w) \div U$

It is evident that in carrying through a given analysis, if, due to any assumptions made, E is larger than the "true" value, these fractions are all increased accordingly and vice versa if E is smaller. In carrying out extensive comparisons of performances of different ships, therefore, the comparisons of the above mentioned quantities containing E will be on the same basis only if the effective horsepowers of the different ships have been arrived at in the same way and based on the same assumptions. Comparisons on such uniform basis should be thoroughly satisfactory even if the values are all somewhat high or low with respect to "true" values. It is also evident that when the history of a given effective horsepower curve is not known it should certainly be viewed with considerable suspicion particularly if comparisons are to be made between trial results.

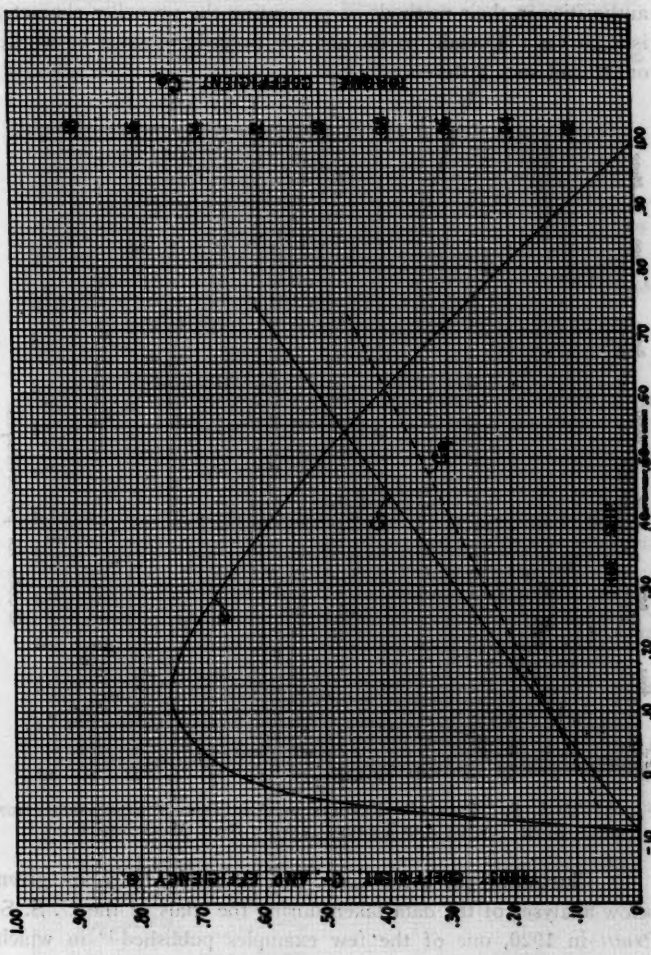


FIG. 1.—PROPELLER CHARACTERISTIC CURVES—U. S. S. "PRUITT."

39. Although the essentials of analyses of trials will be those given above, owing to the different coefficients used by different authorities in their methods of presenting the propeller characteristic curves the details of the analysis procedures must be worked out to suit such differences.

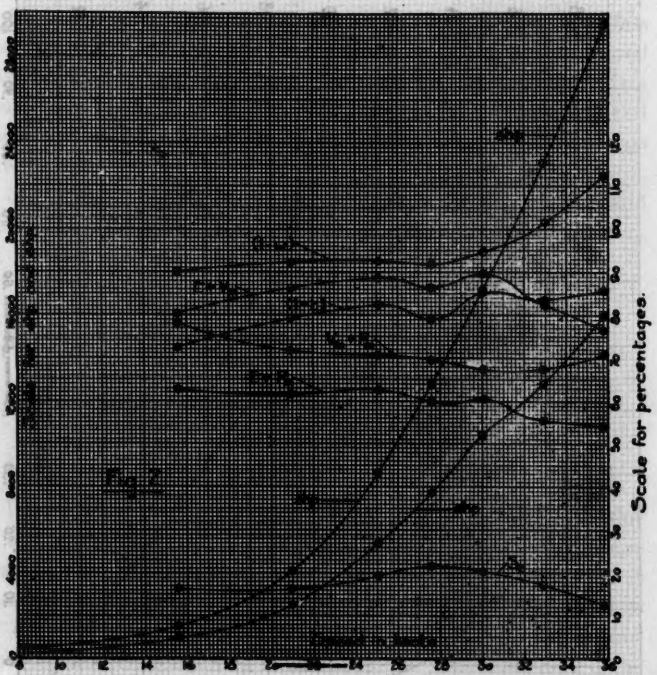


FIG. 2.—U. S. S. "PRUITT." RESULTS OF ANALYSIS OF STANDARDIZATION TRIALS, BOTH TORQUE AND THRUST HAVING BEEN MEASURED, CASE I.

40. Illustrating the three cases, defined above, there are given below analyses of the data taken during the trials of the U. S. S. *Pruitt* in 1920, one of the few examples published[10] in which complete torque and thrust measurements were made.

[10] Propeller Thrust Measurements on U. S. S. *Pruitt*, by John E. Burkhardt. J. A. S. N. E. 1921.

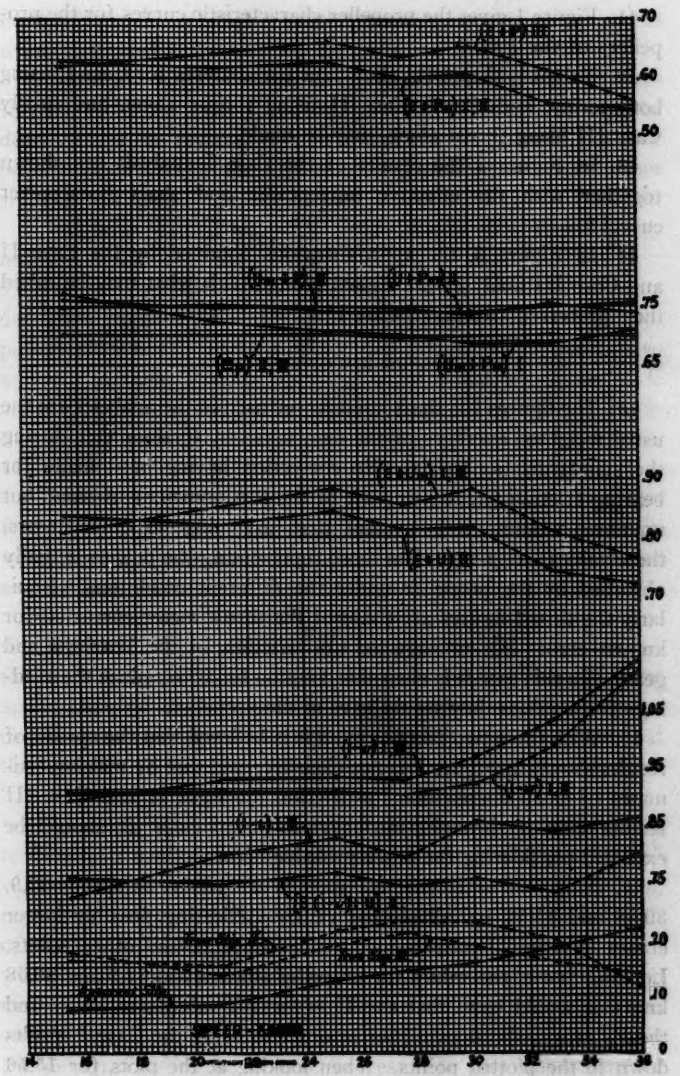


FIG. 3—U. S. S. "PRUITT." COMPARISON OF ANALYSES: CASE I, TORQUE AND THRUST; CASE II, TORQUE ALONE; CASE III, THRUST ALONE.

41. Figure 1 gives the propeller characteristic curves for the propellers of the *Pruitt*.

42. Tables I, II and III show how the analyses for Case I, using both torque and thrust, Case II using torque alone, and finally Case III using thrust alone, may be made.

43. In Figure 2 the results of the Case I analysis are shown together with the effective horsepower and shaft horsepower curves.

44. In Figure 3, the approximations obtained by the Case II and Case III analyses are shown plotted with what may be called the " preferred" results of Case I.

DISCUSSION OF RESULTS.

45. The figures for shaft horsepower used in the analyses are the usual weighted averages of the data taken by torsionmeter during the standardization trials. No correction having been made for bearing friction (in shafting aft of the torsionmeter) of small but unknown amount, it is evident that, assuming no other errors, these values are greater than those representing the powers actually absorbed by the propeller under the different conditions. Similarly the thrust figures are those of the thrust meter corrected for known errors but uncorrected for friction in the bearings and gears. In this case it is clear that friction tends to reduce the readings from values they would have at the propeller.

46. With these two facts in mind it is believed that the spread of results shown in Figure 3 is not excessive and that by making estimates of small corrections for friction the Case II and Case III results may be brought into agreement as close as should be expected considering the other probable errors.

47. It may be noted that the spreads are greater at speeds 32.9, 30.08 and 15.56 for all plotted values, indicating possibly larger errors in original measurements than exist at the other points. Looking first at the values for propeller efficiency at 32.9 and 30.08 knots, it would seem that the U_m (thrust) values are lower and the P_m (torque) values higher in order to bring the Case I results down to the plotted points. Then looking at the plots for 15.56 knots it appears that these conditions are reversed. In this con-

nection it may be noted that the thrust measurements at low speeds were questioned at the time of the trials.

48. The plot of $(1 - w)$ shows an almost constant value of wake fraction from the lower speeds up to about 28 knots and then the decrease and the reversal of sign found characteristic in self-propelled model tests at the higher speeds of destroyers and other ships of fine lines.

It will be noted that $U_m \div P_m$, propeller efficiency, remains very uniform in value for all speeds.

Effective horsepower (E) affects the results of $(1 - t)$ (thrust deduction fraction), $E \div U_m$ (hull efficiency), and $E \div P_m$ (propulsive efficiency). In view of the fact that $U_m \div P_m$ remains so uniform in value as noted above, it appears that the values of E are responsible for the falling off of both $E \div U_m$ and $E \div P_m$ at the higher speeds, although such falling off is not evident in $(1 - t)$.

49. However, too many general conclusions should not be drawn from this single case. It is to be hoped that many more cases in which torque and thrust have been measured will be available for analysis in the future.

50. It may properly be said by those who are looking for "scientific accuracy" that these analyses are objectionable, because (a) the propeller characteristic curves have not yet been proved to be applicable even at low speeds and (b) at higher speeds are almost certainly not truly applicable due to cavitation, with the result that the analyses do not give "true" wake, "true" thrust deduction, "true" propeller efficiency and "true" hull efficiency.

51. In reply it may be pointed out that the object herein is not to develop a method of ship trial analysis which will give "true" values—in view of the assumptions previously explained as being necessary to the method adopted this ideal may be definitely out of reach—but to develop a method of mathematical analysis which will give design factors useful to the designer. By using a design method essentially the reverse of the analysis it is maintained that any deviations which enter the results during analysis are lost in the design, *i.e.*, in the reverse process. In any discussions of results obtained by such analyses, comparisons of wake, thrust

deduction and efficiency values, obtained by the mathematical processes described, with values obtained by physical measurements during tests of models or ships should therefore be made with proper reservations.

52. Using the methods illustrated above, the prospective designer should, of course, analyse all the trials for which he can find reliable data and compile design factors for use when new designs are to be worked upon. With such data available, by a judicious comparison of design factors from trials of ships with information from model tests, he may be confident that he has selected factors which will later be proved correct by the trials of the ship. In this as in other work it goes without saying that increase in skill and judgment come only with experience.

ANAL

1	USS [illegible]		
2	LLSL 510		
3	Displace		
4	[illegible]		
5	[illegible]DA		
6	Blade [illegible]		
7	Average		
8	d = 9.1		
9	p = 10.		
10	V	=	35.85
11	E	=	9000
12	N	=	4567
13	n = N ÷ 60	=	7.46
14	n^2	=	55.7
15	n^3	=	416
16	P_m	=	14756
17	T_m	=	84750
18	$C_Q = P_m \times 97.6\ n^3\ p^5\ d^3$	=	.03255
19	$C_T = T_m \div n^2\ p^4\ d^4$	=	.1659
20	Using C_Q ,	s_q =	.135
21	Using C_T ,	s_t =	.12
22	Mean value	s_t =	.1275
23	$V_A = p\ N\ (1-s) \div 101.33$	=	40.2
24	$U_m = T_m\ V_A \div 326$	=	10450
25	$U_m \div P_m$	=	.709
26	E ÷ U_m	=	.765
27	1-t = R ÷ T_m	=	.859
28	1-w = V_A ÷ V	=	1.122
29	E ÷ P_m	=	.542

CASE I TABLE I

ANALYSIS OF TRIALS WITH TORQUE (SHAFT HORSEPOWER) AND THRUST MEASURED.

USS DEWITT Date 8/19/20 Last Undocked Aug. 2, 1920
LWL 510-4 1/4 B 50' - 8" H 31'-3 3/4 NBC 047
Displacement 1207 tons; Wind NE 2-3 Sea Smooth
2 Propellers 3 blades each; PA (sq.in.) 5225
PA÷DA 0.55 HA÷DA 0.64 Mean width ratio 0.42
Blade thickness at axis 6" Blade thickness ratio .0545
Average data of 2 shafts

d = 9.17 ft. d^2 = 84.1 p ÷ d = 1.1
p = 10.17 ft. p^2 103.5 p^3 = 1053

35.86	32.9	30.08	27.6	25.09	20.98	15.56
8000	6400	5200	5850	2700	1250	475
4507	4021	356	321.4	286.5	229.2	167.2
7.66	6.7	5.93	5.35	4.775	3.82	2.784
50.7	44.9	36.25	28.6	22.8	14.6	7.77
448	301	209	153	109	56.1	21.7
14756	11538	8599	6413	4510	2027	756
84750	76000	66150	57500	42650	24500	13810
.03255	.03765	.04046	.0414	.03905	.0357	.0345
.1658	.1944	.2158	.231	.218	.1928	.2041
.135	.135	.21	.22	.195	.165	.155
.12	.16	.193	.212	.193	.16	.175
.1275	.1725	.2015	.216	.194	.1625	.165
40.2	33.4	28.5	25.3	23.2	19.3	14.0
10450	7790	5780	4460	3040	1450	593
.708	.675	.672	.696	.705	.715	.782
.765	.822	.899	.863	.888	.863	.801
.859	.835	.852	.791	.823	.793	.721
1.122	1.015	0.948	0.917	0.925	0.92	0.90
.542	.555	.605	.60	.626	.617	.627

A

1	USS		
2	LLWL		
3	Disp		
4			
5	FA÷D		
6	Blad		
7	Aver		
8	d =		
9	p =		
10	V (knots)	=	
11	E (horsepower per shaft) effective	=	
12	N (r.p.m.)	=	
13	$n = N \div 60$	=	
14	n^2	=	
15	n^3	=	
16	P_m (horsepower per shaft) measured	=	
17	$C_Q = P_m \times 87.5 \div n^3 p^3 d^2$	=	
18	Using C_Q and p ÷ d in propeller characteristic curves	C_T =	
19		η_t =	
20		e =	
21	$T = C_T n^2 p^2 d^2$	=	
22	$V_A = p N (1-s) \div 101.33$	=	
23	$(1-w) = V_A \div V$	=	
24	$U = TV_A \div 326$	=	
25	$U \div P_m$	=	
26	$E \div U$	=	
27	$E (1-w) \div U$	=	
28	$E \div P_m$	=	
29	T ÷ PA (thrust lbs/in^2 of PA	=	
30	W d N (tip speed ft/min)	=	

CASE II TABLE II

ANALYSIS OF TRIALS WITH TORQUE (SHAFT HORSEPOWER) MEASURED.

USS PRUITT Date 8/18/20 Last Undocked Aug. 2, 1920.
LLWL 510-4 1/4 B 30-6 H 9-3 1/4 NBC 0.47
Displacement 1207 tons; Wind NE 2-3 Sea Smooth
2 Propellers 3 blades each; PA (sq.in.) 5225
PA÷DA 0.55 HA÷DA 0.64 Mean width ratio 0.42
Blade thickness at axis a" Blade thickness ratio .0545
Average data of two shafts

d = 9.17 ft. d^2 = 84.1 p ÷ d = 1.11
p = 10.17 ft. p^2 103.5 p^3 = 1053

35.85	32.9	30.08	27.6	25.09	20.98	15.56
8000	6400	5200	3850	2700	1250	475
459.7	402.1	356	321.4	286.5	229.2	167.2
7.66	6.7	5.93	5.35	4.775	3.82	2.786
58.7	44.9	35.25	28.6	22.8	14.6	7.77
448	301	209	153	109	56.1	21.7
14756	11538	8599	6412	4310	2027	758
.03255	.03785	.04065	.0414	.03905	.0357	.0345
.17	.21	.23	.235	.216	.195	.186
.135	.165	.21	.22	.195	.165	.155
.75	.737	.726	.72	.733	.743	.747
86900	82100	70600	58500	42900	24800	12590
39.9	32.8	28.2	25.2	23.16	19.2	14.19
1.112	.997	.938	.915	.923	.915	.912
10630	8260	6110	4525	3046	1460	548
.72	.716	.711	.706	.707	.72	.725
.753	.775	.851	.851	.886	.858	.847
.857	.775	.796	.777	.814	.784	.79
.542	.585	.605	.60	.626	.617	.627
16.62	15.71	13.5	11.2	8.21	4.75	2.41
13850	11576	10250	9260	8350	6600	4380

1	Us		
2	LL		
3	Di		
4			
5	PA		
6	Bl		
7	Av		
8	d		
9	p		
10	V (knots)	=	
11	E	=	
12	N	=	
13	$n = N \div 60$	=	
14	n^2	=	
15	n^3	=	
16	T_m	=	
17	$C_T = T_m \div n^2 p^2 d^2$	=	
18	Using C_T in C_Q	=	
19	propeller character- s_t	=	
20	istic curves, Fig.1 e	=	
21	$P = C_Q n^3 p^3 d^2 \div 87.5$	=	
22	$V_A = pN(1-s_t) \div 101.33$	=	
23	$1-w = V_A \div V$	=	
24	$U_m = T_m V_A \div 326$	=	
25	$U_m \div P$	=	
26	$E \div U_m$	=	
27	$1-t = R \div T_m$	=	
28	$E \div P$	=	
29	$T_m \div PA$	=	1
30	w d N	=	1

C A S E III TABLE III

ANALYSIS OF TRIALS WITH THRUST MEASURED.

USS PRUITT Date 8/18/20 Last Undocked Aug. 2, 1920
LLWL 310'-4"1/4 B 30'-8" H 9'-3"3/4 NBC 0.47
Displacement 1207 tons; Wind NE 2-3 Sea Smooth
2 Propellers 3 blades each; FA (sq.in.) 5225
FA÷DA 0.55 HA÷DA 0.64 Mean width ratio 0.42
Blade thickness at axis Blade thickness ratio
Average data of two shafts

d = 9.17 ft. d^2 = 84.1 p ÷ d = 1.11

p = 10.17 ft. p^2 103.5 p^3 = 1053

=	35.85	32.9	30.08	27.6	25.09	20.98	15.56
=	8000	6400	5200	3850	2700	1250	475
=	459.7	402.1	356	321.4	286.5	229.2	167.2
=	7.66	6.7	5.93	5.35	4.775	3.82	2.786
=	58.7	44.9	35.25	28.6	22.8	14.6	7.77
=	448	301	209	153	109	56.1	21.7
=	84750	76000	66150	57500	42650	24500	13810
=	.1658	.1944	.2155	.231	.215	.1928	.2041
=	.031	.035	.0385	.0405	.0385	.035	.0365
=	.12	.16	.193	.218	.193	.16	.175
=	.75	.746	.735	.725	.735	.746	.74
=	14070	10670	8150	6270	4250	1987	802
=	40.6	35.9	28.85	25.43	23.22	19.53	13.85
=	1.133	1.05	.96	.922	.926	.921	.89
=	10550	7910	5855	4485	3040	1453	585
=	.75	.741	.719	.715	.715	.731	.73
=	.758	.809	.859	.859	.869	.86	.812
=	.859	.835	.852	.791	.823	.793	.781
=	.569	.60	.638	.614	.635	.629	.598
=	16.22	14.56	12.66	11.0	8.16	4.69	2.64
=	13850	11575	10250	9240	8250	6600	4290

PROTECTIVE COVERING FOR RESERVE FEED WATER TANKS.

By LIEUTENANT B. P. WARD, U. S. N.

Cement wash was used quite extensively for coating the inside walls of reserve feed water tanks on board ships of the Navy until recently. As a protective covering it was never considered to be satisfactory, because it had to be renewed very frequently. Its adherence was poor, which allowed rust to form beneath it. However, no suitable substitute was prescribed, so it was continued in use.

In 1929 the Bureau of Engineering arranged for an investigation of the effect of cement wash on the alkalinity of feed water at the U. S. Naval Engineering Experiment Station. It was believed that the use of cement wash had been an aid in the maintenance of alkalinity of the feed water.

In some tests of this point four specimens of ⅛-inch black sheet iron, each of 1.0 square feet total area, were given two coats of cement wash (two parts Portland cement and one part fire clay) made up to a brushing consistency. The coatings were applied on successive days and were allowed to dry over water, in order to minimize the chances of their cracking. Each specimen was then placed in a 10-gallon crock containing eight gallons of distilled water. The platform on which the crocks were located was rocked two times per minute, which caused a washing action of the water on the specimens. The specimens were submerged to different depths in the various crocks. Samples of water were removed from the crocks at regular intervals and analyzed. The precipitate from the cement wash was analyzed, also. The specimens were kept under the above described conditions for thirty days. The results of the analyses are shown on page 354.

Analysis of Dry Cement Wash Mixture

	Per-cent
Silica	30.08
Iron and aluminum oxides	19.58
Lime	41.67
Magnesium oxide	1.40
Sulphur trioxide	0.90
Ignition loss	6.37

Analysis of Precipitate from Cement Wash After 30 Days

	Per cent
Calcium oxide	54.12
Silica	4.72
Iron and aluminum oxides	2.32
Carbon dioxide	38.84

Analysis of Water Samples Removed from Crocks

	After 10 days	*After 30 days*
Alkalinity, % normal		
Hydroxide	0.10	0.10
Carbonate	0.13	0.13
Calcium oxide, grs./gal.	4.95	6.41
Iron and aluminum oxides, grs./gal.	0.32	0.69
Silica, grs./gals.	2.01	2.70
Sulphur trioxide, grs./gal.	1.40	2.54
Carbon dioxide, grs./gal.	1.64	1.65
Gallons of water per sq. ft. immersed area	6.37	6.37

The chemical composition of the dry cement wash mixture consisted of Ca_2SiO_4, Ca_3SiO_5 and $Ca_3(AlO_3)_2$, and a hydrated aluminum silicate. When water is added to these compounds, they become hydrated and setting occurs. This is especially true of tricalcium silicate, Ca_3SiO_5, which forms gelatinous hydrates that gradually harden and help to cement the other hydrates together.

The analysis of the precipitate found in the crocks indicates that the precipitated compounds are largely calcium carbonate, calcium silicate, silica, and iron and aluminum oxides. Most of the precipitate settled to the bottom of the crocks in a loose amorphous mass, which could readily have been carried along with any water withdrawn from the bottom of the containers. There was also an adherent ring of the precipitated material deposited on the inside of

the crocks at the water level. This precipitate was difficult to remove, and then only by the use of strong acid.

The contamination of boiler water by cement wash was proved by submerging cement wash coated cylinders in a small experimental boiler operating with distilled water in a closed system and using no make-up feed. The cement wash went into solution and was deposited as a scale on the boiler tubes.

Both the hydrozyl and carbonate alkalinity of the water remained constant after the first week of the test. This alkalinity was due to the compounds of calcium hydroxide and calcium carbonate in the water. The small amount of calcium hydroxide found in the water was due both to its slow rate of dissolving and to the fact that it would react with any bicarbonates which are formed when carbon dioxide gas dissolves in water to form normal carbonates. This latter reaction tends to prevent its accumulation. The amount of calcium carbonate dissolved in the water represents a saturated condition, which accounts for its constant value during the test.

The water analysis shows that the compounds dissolved in the water were principally calcium carbonate, calcium sulphate, calcium silicate, and iron and aluminum oxides. The presence of any of these compounds in a boiler feed water is undesirable, because the calcium carbonate forms a soft scale in feed heaters, feed pumps, and feed line piping, while the other compounds form scale in the boilers, especially calcium sulphate and calcium silicate which form hard, adherent scales.

The cement wash coatings on specimens which were partially exposed to the air did not hold up as long as those parts of the coatings which were totally submerged. The alternate exposure of a portion of these coatings to water and then to air weakened their resistance to erosion.

The tests showed: first, that boiler feed water stored in tanks coated with cement wash becomes slightly alkaline due to the solution of the basic compounds of calcium from the cement wash; second that, due to solution and disintegration of the coatings, practically all of the materials in the cement wash eventually are carried by the feed water into the boilers where, under boiler conditions, these materials form hard, adherent scale; and, finally,

that to keep the tanks fully protected from corrosion, it would be necessary to renew the coatings of cement wash about four times a year.

These results started a search for a protective covering for feed water tanks that would inhibit corrosion and not contaminate the water. Paints and combinations of paints were obtained from practically every known source of production of protective coatings in this country.

After several experiments were conducted for the purpose of developing a standard and fair method of test, which would simulate service conditions, the tests were made by applying the coverings, in accordance with the directions of the manufacturers, to both sides of black iron test plates, each $8 \times 9 \times \frac{1}{8}$ inches, wire-brushed for removal of loose scale, rust, and dirt, and then submerging the plates in two baths of water for a period of thirty days. One bath consisted of fresh water condensate, which contained less than 0.3 grain of chlorine per gallon. It was heated to 170 degrees F. and held at that temperature for eight hours a day, and then allowed to cool during the night. The second bath was made of the same condensate treated to contain 0.5 per cent normal alkalinity and 25 grains per gallon salinity. It was heated to 120 degrees F. and held at that temperature for eight hours a day and then allowed to cool at night. The coated plates were suspended from a horizontal rack of an oscillating mechanism so as to hang partially submerged in the test tanks. The plate-holding rack was driven at four cycles per minute, imparting to the plates a forward and downward movement through the water during half of the cycle and an upward and backward movement during the second half of the cycle. The plate surfaces remained perpendicular to the plane of the water level during the complete cycle.

Sixty-four samples of coverings, which included paints, combinations of paints, primers, varnishes, oils, and sprayed metal, have been tested. The treated bath at 120 degrees F. was slightly more damaging to the coverings than the untreated bath at 170 degrees F. Several of the coverings have completely prevented corrosion, but their surfaces softened to the extent that slight abrasion removed the covering and exposed the bare plate. Others

failed completely. Only one covering has withstood the conditions of the test entirely satisfactorily. That covering consists of a red lead-chromate primer mixed in a specially flexible vehicle, and an outer coating of aluminum varnish. It allowed no corrosion to form on the plate and its surface remained hard and smooth.

The investigation has not been concluded. Additional tests are being made using different methods of application of the covering which was found satisfactory, and also of other materials that promise protective value.

DISCUSSION.

THE TRAGEDY OF *R–101*.

By E. F. Spanner.

Although it may be unusual for an author to comment upon the criticisms of a reviewer, I am pleased with the opportunity to make the following remarks in regard to the review of "The Tragedy of *R–101*" which appeared in this JOURNAL for February, 1932.

I am not an "unbeliever in the practicability of airships." My view is that while rigid airships have no commercial future as large passenger carrying vehicles, they can be made of value, for mail carrying, in much smaller sizes. It should not be overlooked that while I gave detailed reasons for my objection to the designs of *R–100* and *R–101*, I also put it on record that I anticipated that the *Graf Zeppelin* would achieve a technical success. I was correct in my judgments, but I have yet to learn that the *Graf Zeppelin* has proved large rigid airships to be practical commercial propositions for carrying passengers.

With regard to my reconstruction of the disaster, Mr. Burgess is entirely wrong in saying that, with the assumptions I have made "the ship would have tended to break in the direction opposite to that assumed by Mr. Spanner." Figure 1 shows the "Earthward Plunge of *R–101*" as I have reconstructed it. It is perfectly plain from this diagram—which, incidentally, agrees with the evidence—that my assumptions do not contradict one another. There is an excess downward pressure on the nose, and an excess downward pressure on the tail at the moment of collapse.

The point upon which Mr. Burgess' argument founders is in relation to the alleged "recovery" of the airship. The actual evidence is overwhelmingly against any idea that there was any "recovery" of the *R–101*. The "apparent recovery" recorded by Mr. Leech, and referred to by Mr. Burgess, is fully in line with Figure 1, and is easily explained by elementary dynamics.

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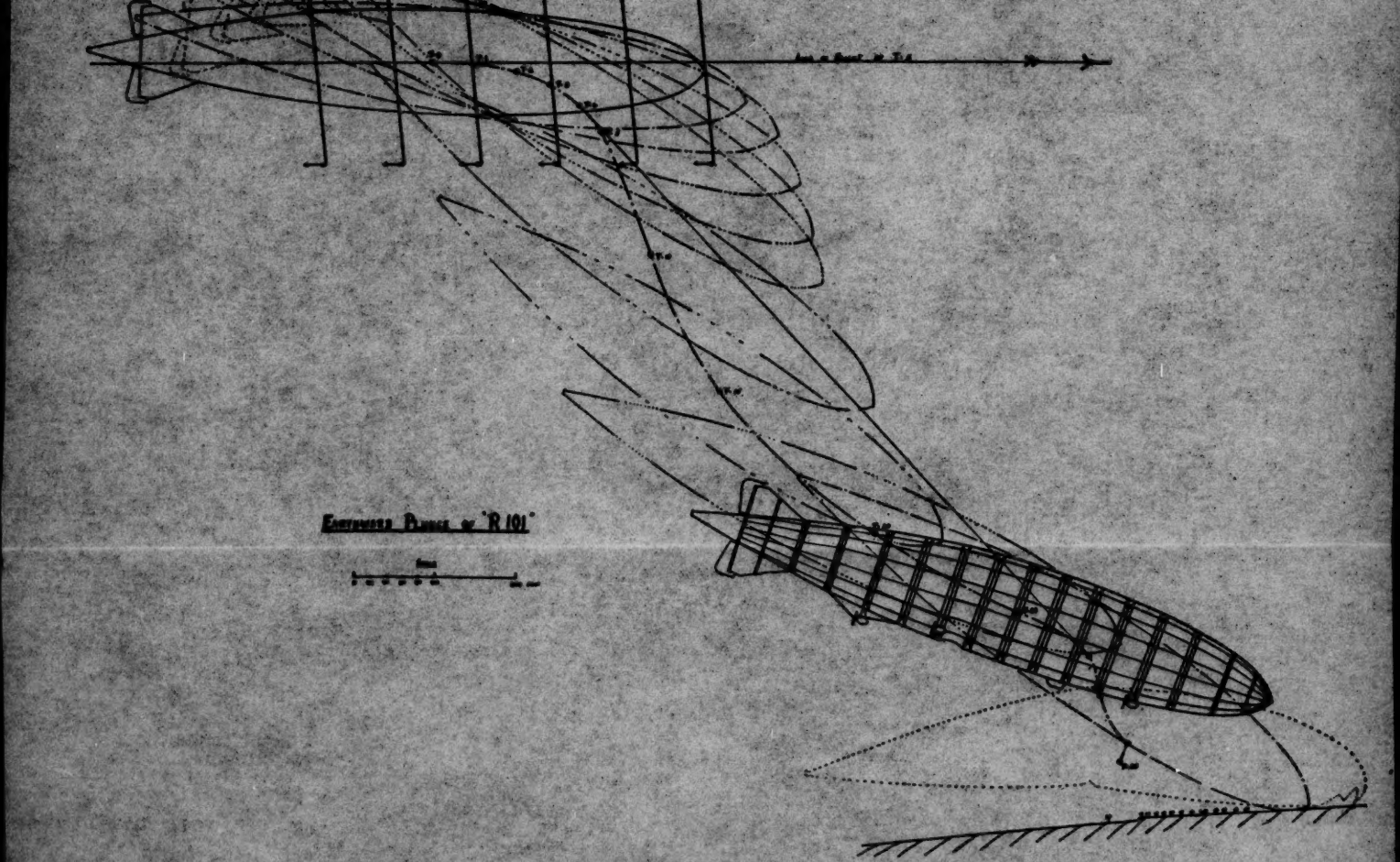

FIGURE 1.

If Mr. Burgess will study Major Scott's I. N. A. paper of 1931, and Dr. Simpson's evidence at the Inquiry, he will find that my estimate of a 66 m.p.h. gust is perfectly in order. I have no information in regard to the tests on the *Los Angeles*, but there is strong corroborative evidence of the instantaneous effect of gusts, from the mast behavior of the *R–101* herself. Mr. Burgess is extremely reluctant to accept this particular idea. I am sorry about this, for I am convinced that gusts are much more serious in their effect than airship designers are prepared to admit.

Mr. Burgess states definitely that "The airship factor of safety of 2 is based on the yield point of the material." That remark may be correct as applied to the calculations for the ships constructed to Mr. Burgess' designs. It is quite incorrect as applied to the case of *R–101*. In the *R–101* the factor of safety of 2 was, in many of the calculations, based upon the ultimate strength, and *not* upon the yield point. Further the calculations, which were handled differently from those for which Mr. Burgess is responsible, were unsatisfactory for a vessel of the type of construction used in *R–101*.

The above are simply points of argument upon which Mr. Burgess may modify his opinion after further study. In the meantime the points are worthy of note and study. In preparing Figure 1, I made certain assumptions, my reasons for which I gave—chapter and verse. It is a little misleading of Mr. Burgess to suggest that I lay claim to omniscience in making such assumptions. After all, Mr. Burgess must recognize that one is at liberty to make any reasonable assumption in reconstructing a disaster. The test comes when the conclusions arising from these assumptions are tested against recorded facts.

Mr. Burgess again runs the risk of discrediting my work when he winds up his review by saying "Mr. Spanner naively states that he had formed his theory long before he obtained the evidence of the wreck." This is a misleading half-truth. Mr. Burgess should have completed the remark by adding, "It must be admitted, however, that Mr. Spanner read and digested nearly 500,000 words of evidence and argument before arriving at his theory."

Figures 2 and 3 show how closely my ideas correspond with the recorded facts.

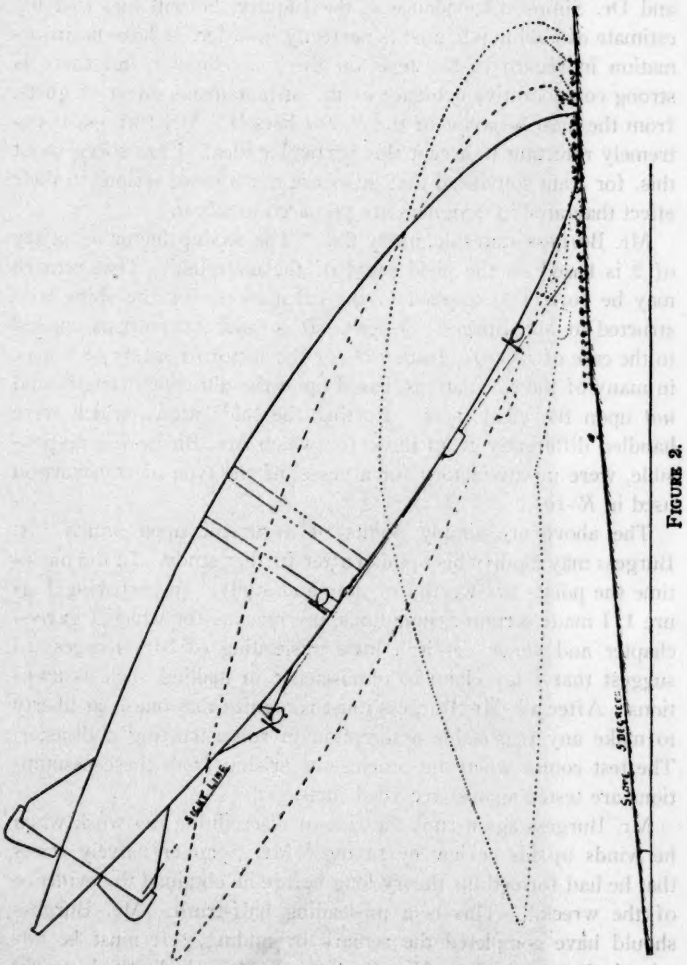

FIGURE 2.

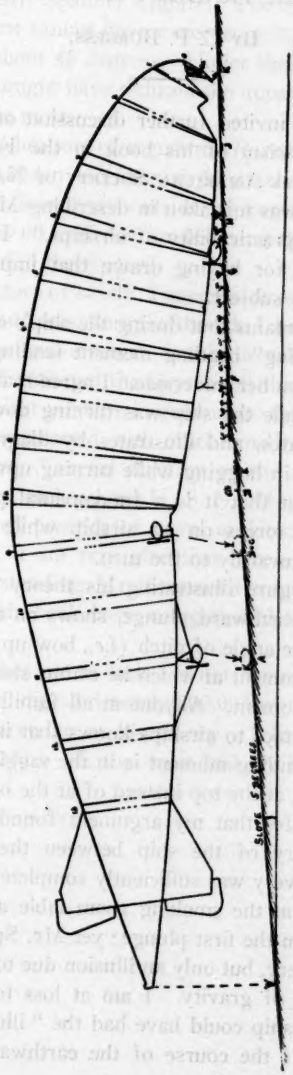

FIGURE 3.

THE TRAGEDY OF *R-101*.

By C. P. Burgess.

Mr. Spanner has invited further discussion on this subject by replying to my criticism of his book in the February issue of The Journal of the American Society of Naval Engineers.

I am glad that I was mistaken in describing Mr. Spanner as an "unbeliever in the practicability of airships." I take this opportunity to apologize for having drawn that impression from his several books on the subject.

Mr. Spanner maintains that during the ship's earthward plunge, there was a "hogging" bending moment tending to crumple the ship longitudinally on her underside. I agree that the bending was in that direction while the ship was turning downward; but Mr. Spanner clearly states, and illustrates by diagrams, his opinion that the ship broke in hogging while turning upward. I can only repeat my statement that it is a fundamental principle of aerodynamics that the forces on an airship while turning produce bending moments inwardly to the turn.

Mr. Spanner's figure illustrating his theory of the course of events during the earthward plunge, shows an upward turn with the ship at a positive angle of pitch (*i.e.*, bow up) *relatively to the flight path* at the moment at which he claims she was broken by a hogging bending moment. Anyone at all familiar with the application of aerodynamics to airships knows that in such an attitude the aerodynamic bending moment is in the sagging direction, producing compression at the top instead of at the bottom of the ship.

Mr. Spanner states that my argument founders in relation to the alleged recovery of the ship between the initial and final plunges. The recovery was sufficiently complete to permit picking up and replacing on the smoking room table articles which had rolled to the floor in the first plunge; yet Mr. Spanner claims that there was no recovery, but only an illusion due to centrifugal force masking the effect of gravity. I am at loss to understand how anyone within the ship could have had the "illusion" of a return to the horizontal if the course of the earthward plunge was in

accordance with Mr. Spanner's figure. The centrifugal force due to the upward turn cannot have exceeded g/4, and the inclination of the ship was about 45 degrees. Under these circumstances the centrifugal force might have reduced the apparent inclination by 8 or 9 degrees at the most.

I do not know what sort of a gust the *R–101* encountered; but I maintain that a sudden increase of 44 m.p.h. in the relative air speed of the ship is entirely outside of experience in rough air flying, and there is no evidence for believing that such an improbable occurrence befell the *R–101*.

In regard to factors of safety, I agree with Mr. Spanner that the design of the *R–101* was faulty, although I believe she had sufficient primary structural strength for the weather she encountered that fatal night. I gathered the impression that Mr. Spanner was making a case against airships in general on the grounds that they are customarily constructed with much smaller factors of safety than other vehicles. Careful scrutiny of the design assumptions for recent airships, such as the U. S. S. *Akron*, and comparison with the established practice of naval architecture shows similar factors of safety. I am very glad that here again the error was mine in misunderstanding Mr. Spanner's meaning. I hope that other readers of his book have not gathered the same erroneous impression.

Mr. Spanner takes me to task for not having given him credit for having read and digested nearly 500,000 words of evidence and argument before arriving at his theory. I hasten to say that I admire and sympathize. I, too, have had my labors on the same subject.

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The Measurement of Noise.
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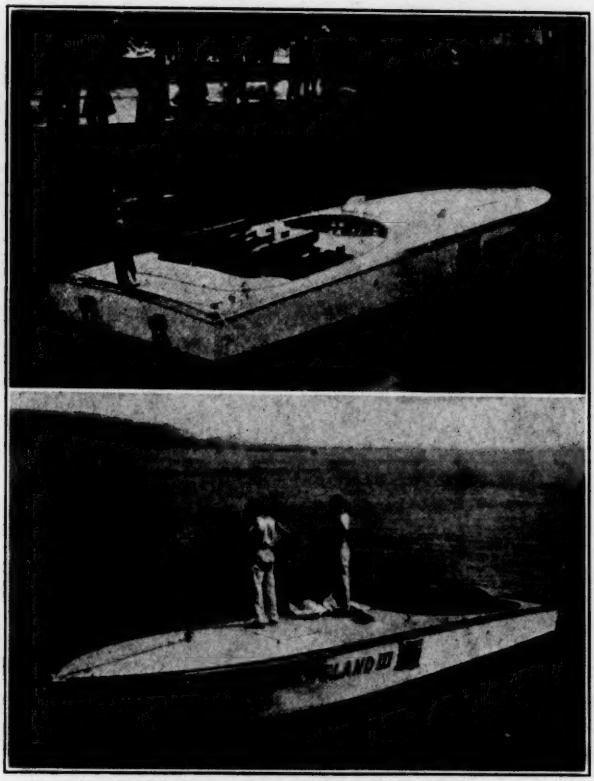


"Miss England III."

MISS ENGLAND III.

Marine Propulsion Extraordinary.

The launch from the Thornycroft works at Hampton-on-Thames, just over a week ago, of Lord Wakefield's motor speed-boat *Miss England III* serves to show what tremendous strides have been made in marine propulsion in recent years. In a single-step hull of light but strong construction, only 35 feet long by 9 feet 6 inches broad, 3950 B.H.P. have been crammed, and the specific weight of hull and machinery is only 3.23 pounds per B.H.P. Propulsion is by means of two vee-type 12-cylinder Rolls-Royce supercharged racing aeroplane engines, such as were used in the last Schneider Trophy race. Unlike *Miss England II*, twin screws are used and, of particular interest, they rotate at a slower speed than did the single propeller of the earlier boat—7000 as against 12,000 R.P.M. The screws are of the two-bladed type, and the speed-increasing gear-boxes are located forward of the engines and cockpit, which will be occupied by the intrepid Mr. Kaye Don. It is hoped to attain a speed of 130 miles per hour on Lake Garda during this month with this truly remarkable British craft.

Designed and built as a new challenger for the Harmsworth Trophy races to be held at Detroit in the autumn.—"Shipbuilding and Shipping Record," May 19 and 26, 1932.

A news dispatch from Loch Lomond, Scotland, dated July 19th, states that Kaye Don established a record of 119.81 miles per hour with *Miss England III*.

MISS AMERICA X.

For comparison with the data given for *Miss England III*, the following information about Mr. Gar Wood's defenders was given in an Associated Press dispatch of July 16: *Miss America X* is 38 feet long with a maximum beam of 10 feet 6 inches; her power plant consists of two 12-cylinder engines in tandem on each of the two shafts, total horsepower being 6400 for the twin screws. *Miss America IX*, holding a straight-away record of 111.712 miles an hour, is 10 feet shorter, is 4 inches less in beam and has 2800 horsepower.

THE PRACTICAL ASPECT OF CREEP.

From a consideration of the mass of experimental data which is rapidly being collected concerning the creep effect to be found in materials subjected to load at elevated temperature, it would seem that ideas on the present basic conceptions as to the physical properties of engineering materials are about to be radically altered. It has been known for many years that the strength of metals is considerably reduced at high temperature; but the research carried out in recent years has shown that this reduction of strength is not merely one of amount, but also of kind. It is certain that with the materials which have been investigated there is a critical temperature above which the materials, while retaining considerable rigidity, nevertheless assume a plastic state in which almost any load will cause deformation, the rate of such deformation depending on the value of the temperature and the intensity of the stress. From the point of view of the engineer it is necessary to have some basis for safe and satisfactory working, and although many cases may be met at present by the provision of special steels in which the creep is negligibly small at reduced working stress, there is little doubt that future development will make a better understanding of the mechanism of creep

very desirable. Since creep generally manifests itself in a preliminary relatively high rate, which settles down to a slow and measured rate of extension, it is probable that one line of attack will be to produce a material of which the creep strain is asymptotic to some limiting and determinable value upon which calculations can be based. The alternative to this is to include the life of a part in the calculations for its strength, in order that creep will not become excessive before renewal takes place.—"Mechanical World," March 25, 1932.

A COMPARISON OF RECENT STEAM TABLES.

By P. A. Willis, G. A. Hawkins and A. A. Potter.*

During the past ten or twelve years marked increases have taken place in the pressures and temperatures used in steam power generation. In 1920 the maximum pressure in power station practice was 300 pounds per square inch. This was raised to 350 pounds per square inch in 1923, and the following year plants designed for 650 pounds were under construction. By 1926 the maximum steam pressure had been raised to 1200 pounds per square inch and there are now several central stations operating with pressures of about 1400 pounds. During the past four or five years special types of high-pressure steam generators have been under development both in this country and in Europe, some of these being designed to produce steam at pressures above the critical.

Steam temperatures have increased from about 600 degrees F. in 1920 to over 800 degrees at the present time, with a few European stations operating at 900 degrees and one large public utility in this country has an experimental installation designed to generate and use steam at 1000 degrees.

NEED FOR NEW STEAM TABLES.

The steam tables of ten years ago were satisfactory for pressures up to about 350 pounds per square inch. A few of these tables covered the entire range of pressures up to the critical, but the lack of experimental data at the higher pressures made for a considerable amount of disagreement and uncertainty. The rapid increases in pressures and temperatures for a time outstripped the progress in steam research, and designers of steam power equipment found themselves without adequate data covering the fundamental properties of steam.

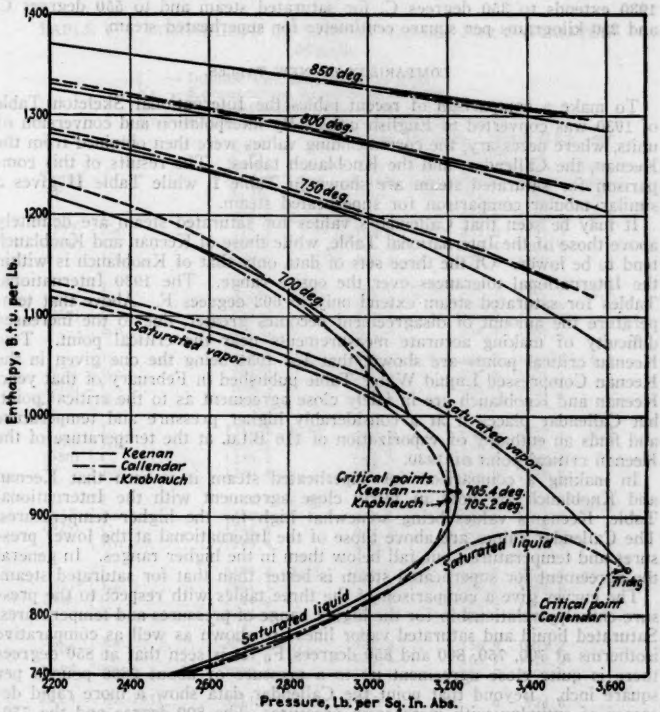
To remedy this situation the American Society of Mechanical Engineers undertook, in 1921, a definite program of steam research. At about the same time Prof. H. L. Callendar[1], in England, began a research project covering the higher ranges of pressures and temperatures. He was aided by the British Electrical and Allied Industries. In Germany a research program of similar scope has been in progress for a number of years. It has been promoted by the Verein deutscher Ingenieure and the Notgemeinshaft der deutschen Wissenschaft and has been mainly under the direction of Dr. Ing. Knoblauch.

*Purdue University.

[1]Since Professor Callendar's death in January, 1930, his research is being carried on by Professor Edgerton assisted by the former's son, G. S. Callendar, who is doing the experimental work.

WORK ON NEW STEAM TABLES IN THE UNITED STATES AND IN EUROPE.

Beginning in 1922 the steam research work in the United States was carried on simultaneously at three institutions: the United States Bureau of Standards, Massachusetts Institute of Technology and Harvard University. The experimental work at Harvard was completed by 1925, but the work at the other two institutions is still in progress. In 1927, using the data available by that time, Prof. J. H. Keenan undertook the work of formulating a steam table to extend to 3500 pounds and 1000 degrees. The work was completed in 1929 and resulted in the Keenan Steam Tables and Mollier



COMPARISON OF KEENAN, CALLENDAR AND KNOBLAUCH VALUES FOR ENTHALPY IN REGION OF CRITICAL POINT.

Diagram which was published in 1930. In 1929 Prof. Callendar published his "Extended Steam Tables" covering pressures up to 4000 pounds and temperatures up to 850 degrees. The latest steam tables to come from Germany are those of Knoblauch, Raisch, Hausen and Koch. They were published early in 1932 and extend to 250 kilograms per square centimeter (3556 pounds per square inch) and 550 degrees C. (1022 degrees F.).

In order to secure better coordination of research activities in the different countries an international steam table conference was held in London in 1929. At this meeting a skeleton steam table was made up, consisting of what appeared to be the most probable values at that time. Plus or minus tolerances were inserted in this table, it being agreed that any investigator's data which came within the tolerance for that portion of the table would be considered worthy of recognition.

A second international conference was held at Berlin in 1930, at which time the skeleton table was revised in accordance with the progress made since the previous meeting. It is expected that a third conference will be held in this country early next fall. The International Skeleton Table of 1930 extends to 350 degrees C. for saturated steam and to 550 degrees C. and 250 kilograms per square centimeter for superheated steam.

COMPARISON OF NEW TABLES.

To make a comparison of recent tables the International Skeleton Table of 1930 was converted to English units. By interpolation and conversion of units, where necessary, the corresponding values were then obtained from the Keenan, the Callendar, and the Knoblauch tables. The results of this comparison for saturated steam are shown in Table I, while Table II gives a similar tabular comparison for superheated steam.

It may be seen that Callendar's values for saturated steam are definitely above those of the International Table, while those of Keenan and Knoblauch tend to be lower. Of the three sets of data only that of Knoblauch is within the International tolerances over the entire range. The 1930 International Tables for saturated steam extend only to 662 degrees F. Above that temperature the amount of disagreement becomes greater, due to the increased difficulty of making accurate measurements near the critical point. Two Keenan critical points are shown, that for 1931 being the one given in the Keenan Compressed Liquid Water Table published in February of that year. Keenan and Knoblauch are in fairly close agreement as to the critical point, but Callendar places it at a considerably higher pressure and temperature and finds an enthalpy of vaporization of 116 B.t.u. at the temperature of the Keenan critical point of 1930.

In making a comparison for superheated steam it is seen that Keenan and Knoblauch are both in fairly close agreement with the International Table, Keenan's values being somewhat high for the higher temperatures. The Callendar values are above those of the International at the lower pressures and temperatures, but fall below them in the higher ranges. In general the agreement for superheated steam is better than that for saturated steam.

The curves give a comparison of the three tables with respect to the pressure-enthalpy relationship for the higher range of pressures and temperatures. Saturated liquid and saturated vapor lines are shown as well as comparative isotherms at 700, 750, 800 and 850 degrees F. It is seen that at 850 degrees there is quite close agreement up to a pressure of about 2800 pounds per square inch. Beyond that point the Callendar data show a more rapid decrease of enthalpy with increase in pressure. The 800-degree and the 750-degree isotherms show the Callendar values above those of Keenan up to about 2800 pounds, but again with a more rapid decrease at the higher pressures. At 3500 pounds per square inch and 750 degrees Callendar's value for enthalpy is about 40 B.t.u. below that of Keenan.

At 700 degrees Keenan and Callendar agree up to about 2500 pounds per square inch. Keenan's isotherm then proceeds with a greater amount of curvature which throws it above Callendar's most of the remaining distance to the saturation line. The Knoblauch 700-degree isotherm is considerably

above those of both Keenan and Callendar up to about 2800 pounds. It then falls at a more rapid rate than either of the other two.

The saturation lines of Keenan and Knoblauch agree reasonably well and their critical points are fairly close together with only two-tenths of a degree difference in temperature and 25 pounds difference in pressure. Callendar is in distinct disagreement, placing his critical point at 3650 pounds per square inch and 717 degrees F. In addition, Callendar reverses the curvature of the saturated liquid line beyond 3200 pounds so that the two curves become tangent at the critical point instead of meeting to form a smooth curve of parabolic form as do those of Keenan and Knoblauch.

TABLE I — COMPARISON OF RECENT TABLES FOR SATURATED STEAM

I — International mean value — 1930.
Ke — Keenan tables, interpolated — 1930.
C — Callendar tables, interpolated — 1929.
Kn — Knoblauch, Raisch, Hausen & Koch — 1932.

Temp. F.		Abs. Press. Lb. per Sq.In.	Int. Toler. ±	Total Enthalpy. Mean B.t.u. per Pound Sat. Liquid (h_f)	Int. Toler. ±	Sat. Evap. (h_{fg})	Sat. Vapor (h_g)	Int. Toler. ±
392 (200)	I.	225.6	0.14	366.2	0.18	833.9	1200.1	4.5
	Ke.	225.5		366.3		833.0	1199.3	
	C.	227.4		366.5		838.4	1204.9	
	Kn.	225.5		366.2		833.2	1199.4	
482 (250)	I.	557.5	1.42	466.4	0.90	737.3	1203.7	7.2
	Ke.	577.1		467.4		735.2	1202.6	
	C.	582.1		467.5		744.7	1212.2	
	Kn.	576.7		466.4		734.7	1201.1	
527 (275)	I.	863.3	1.42	520.0	1.8	678.3	1198.3	9.0
	Ke.	863.2		523.1		671.5	1194.6	
	C.	868.5		522.0		685.3	1207.3	
	Kn.	862.3		519.8		674.4	1194.2	
572 (300)	I.	1247.4	1.42	579.4	3.6	602.7	1182.1	9.0
	Ke.	1247.2		583.2		596.1	1179.3	
	C.	1249.0		580.3		613.7	1194.0	
	Kn.	1246.4		577.9		601.9	1179.8	
617 (325)	I.	1749.4	1.42	647.7	5.4	509.2	1156.9	10.8
	Ke.	1750.6		648.7		506.0	1154.7	
	C.	1744.1		645.3		522.3	1167.6	
	Kn.	1749.4		644.1		509.2	1153.3	
662* (350)	I.	2399.4	2.13	726.9	9.0	379.7	1106.6	14.4
	Ke.	2401.3		729.6		376.0	1105.6	
	C.	2384.0		724.0		392.7	1116.7	
	Kn.	2398.6		723.3		376.1	1099.4	
700	Ke.	3096.4	....	846.3	....	156.9	1003.2	....
698	C.	3099.0	....	833.5	....	188.7	1022.2	....
	Kn.	3050.4	....	824.1	....	181.7	1005.8	....
705.4	Ke.-1931.	3211.6	....	924.8	....	0	924.8	....
706.1	Ke.-1930.	3226.0	....	925.0	....	0	925.0	....
	C.	3246.3		875.8		116.1	991.9	
705.2*	Kn.	3201.0	....	915.8	....	0	915.8	....
710	C.	3350.0	....	883.2	....	84.3	967.5	....
717	C.	3650.0	....	846.0	....	0	846.0	....

* Upper limit of 1930 International tables.
Temperatures in parentheses are in degrees centigrade.

TABLE II — COMPARISON OF RECENT TABLES FOR SUPERHEATED STEAM (Mean B.t.u. per pound)

I — International mean value — 1930.
t — International tolerance (plus or minus)
Ke — Keenan tables, interpolated — 1930
C — Callendar tables, interpolated — 1929
Kn — Knoblauch, Raisch, Hausen & Koch — 1932
Temperature — degrees F.

Pressure Lb. per Sq.In.		482 (250)	572 (300)	662 (350)	752 (400)	842 (450)	932 (500)	1022 (550)
355.6 (25)	I.	1239.7	1294.6	1345.0	1394.5	1442.1	1488.9	1536.6
	t.	3.6	3.6	3.6	3.6	3.6	4.5	5.4
	Ke.	1238.3	1293.1	1343.7	1393.4	1442.2	1490.8	1538*
	C.	1246.6	1299.8	1350.0	1397.3	1444.1	1489.7	.....
	Kn.	1238.5	1293.5	1344.2	1393.7	1442.5	1490.9	1539.5
711.2 (50)	I.	.....	1259.5	1320.7	1376.5	1427.7	1479.9	1530.3
	t.	.....	5.4	4.5	4.5	4.5	4.5	6.3
	Ke.	.....	1258.4	1319.8	1375.7	1428.9	1481.2	1533*
	C.	.....	1264.4	1324.7	1378.8	1429.3	1477.6	.....
	Kn.	.....	1259.0	1321.9	1377.4	1430.6	1482.3	1535.3
1422.3 (100)	I.	.....	.....	1262.2	1336.0	1399.0	1457.4	1513.2
	t.	.....	.....	5.4	4.5	4.5	4.5	7.2
	Ke.	.....	.....	1260.4	1333.9	1399.9	1461.7	1520*
	C.	.....	.....	1266.3	1339.6	1400.2	1454.9	.....
	Kn.	.....	.....	1265.0	1338.0	1401.5	1460.3	1516.8
2133.5 (150)	I.	.....	.....	1166.8	1286.5	1365.7	1433.1	1494.3
	t.	.....	.....	7.2	4.5	4.5	4.5	7.2
	Ke.	.....	.....	1167.3	1283.0	1368.2	1441.6	1507*
	C.	.....	.....	1166.2	1289.0	1367.1	1430.7	1487*
	Kn.	.....	.....	1166.8	1287.7	1368.2	1434.0	1495.8
2845.0 (200)	I.	.....	.....	.....	1216.3	1326.1	1407.1	1475.4
	t.	.....	.....	.....	4.5	5.4	7.2	10.8
	Ke.	.....	.....	.....	1219.4	1330.6	1420.3	1493*
	C.	.....	.....	.....	1215.8	1326.8	1403.6	1466*
	Kn.	.....	.....	.....	1222.6	1329.9	1405.3	1473.4
3556.0 (250)	I.	.....	.....	.....	1121.0	1277.5	1376.5	1452.0
	t.	.....	.....	.....	7.2	9.0	9.0	18.0
	Ke.	.....	.....	.....	1126*	1289*	1396*	1478*
	C.	.....	.....	.....	1091.0	1275.3	1374.8	1461*
	Kn.	.....	.....	.....	1124.4	1284.0	1373.4	1448.4

* Extrapolated values.
Pressures in parentheses are in kilograms per square centimeter
Temperatures in parentheses are in degrees centigrade.

The critical point is usually considered as that pressure and temperature at which the meniscus disappears when a liquid is confined in a transparent tube and heated. Callendar found this phenomenon to occur at about the same temperature and pressure as is shown by the critical points of Keenan and Knoblauch, but claimed to be able to detect a difference in density of the liquid and the vapor above that point on up to the critical point which he indicates. He later made direct calorimetric measurements of enthalpy in this region which, he claimed, verified his first observations.

Much valuable research in steam properties has been completed in recent years and further search for new knowledge in this field is still in progress. Better agreement among the different investigators is being brought about each year and it may be expected that within a short period of time no differences of importance will exist among the several tables. The data now available appear to be amply accurate for present requirements in steam power practice. Further increases in pressures and temperatures will no doubt demand a somewhat greater refinement in steam table values.—"Power," June 7, 1932.

EFFECT OF HUMIDITY ON POWER OF INTERNAL-COMBUSTION ENGINE

The power which can be developed in an automotive engine is proportional to its mass rate of oxygen consumption.

This statement summarizes the results of research at the Bureau of Standards during the last 15 years and furnishes an interesting contrast to the complex formulas developed to account for the effects of various atmospheric factors on engine performance. This simple formula materialized when results of the final research, that of the effect of humidity, were considered together with results of earlier work, on the effects of other atmospheric factors on engine performance.

Previous work had verified the quotation above over wide ranges of temperature and air pressure. The latest work, which will be discussed in a Technical Report of the National Advisory Committee for Aeronautics, has proved the truth of the summary over a wide range of humidity.

Although all of this work was done on the grounds of the bureau in Washington, the atmospheric conditions were artificially varied over such ranges that it is now possible to predict the performance of an engine anywhere on the surface of the earth—in the hot, dense atmosphere of Death Valley; in the oppressive humidity of the tropics; or in the thin, cold air at the peak of Mount Everest.

The bureau's altitude laboratory, in which a portion of this work was done, is a testing chamber of such versatility that the entire sequence of experiments could have been made therein. The greater portion of the research was carried out, as a matter of convenience, in the automotive laboratory, where special apparatus was designed and constructed to enable the artificial control of atmospheric conditions. It was also necessary to develop a new type of psychrometer, capable of precise measurement of humidity over the extreme range used. This instrument will be described later in a Bureau of Standards publication.—"Technical News Bulletin," Bureau of Standards, March, 1932.

THE PRATT & WHITNEY TWIN WASP JUNIOR ENGINE.

By Joseph E. Lowes, Jr.*

The Twin Wasp Junior, fourteen-cylinder, two-row, air-cooled, radial engine, designed and manufactured for the United States Navy, by the Pratt & Whitney Aircraft Company, marks the first departure of the company from the fundamental design of the single-row Wasp, which since 1925 has held a predominant position in the aircraft field.

The Twin Wasp Junior was conceived with the idea of decreasing head resistance for high speed aircraft and to improve vision over that possible with a single row of cylinders for an engine of the same displacement, the general specifications for the engine having been laid down by the Bureau of Aeronautics, Navy Department.

The engine consists of two banks, having seven cylinders in each, the individual banks resembling closely and incorporating the general features of the single-row models. The cylinder design follows standard Pratt & Whitney practice of using cast cylinder heads screwed onto finned steel cylinder barrels. The cylinders have integrally cast rocker boxes and a special arrangement of closely spaced cooling fins as first introduced on the

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*Pratt & Whitney Aircraft Company.

Wasp Junior single-row engines. The main crankcase is built in three sections, designed to be forged, but the first engines have heat treated cast cases. These will be forged in production. The center section represents half of the main case for front and rear banks, while the front and rear main sections are alike and complete the main case assembly. These sections are bolted together in the manner employed on other Pratt & Whitney models. The front and rear main case sections carry the cams and tappets from whence push rods extend to the valve rockers for front and rear banks of cylinders respectively. The front crankcase section, which carries the thrust bearing and support for the front end of the crankshaft, is fastened to the tappet extension of the front main case on both geared and direct-drive engines. This section is fitted with a built-in valve for the operation of the Hamilton Standard hydro-controllable pitch propeller, which makes possible improved take-off and climb. It is also provided with separate scavenging means for the return of oil to the tank during a steep dive.

The rear of the engine consists of a blower section and rear of similar design to those used on all other Pratt & Whitney engines.

The crankshaft is constructed of one forged piece having the crank throws set at 180 degrees to each other and supported on four bearings. The master connecting rods are of special two-piece construction and carry steel back bronze lined bearing shells which operate on hardened crank pins. The articulated rods are similar to those of the Wasp Junior and have bronze bushings for attachment to knuckle pins and piston pins. Master and knuckle pin bearings are pressure lubricated in the usual manner.

Pistons are of forged aluminum alloy of special Pratt & Whitney design, each having three compression rings and one scraper. The inlet valves are of Tungsten steel and the exhausts chrome nickel steel, both being mounted in bronze guides and seating on forged aluminum bronze seats. The exhaust valves are internally cooled with sodium.

There is a triple oil pump assembly, mounted on the rear section, comprised of two scavenging pumps and one pressure pump. The oil supply for the main bearings is introduced into the front end of the crankshaft. One scavenging pump empties the sump and the other the front crankcase section.

The propeller hub is of the No. 30 S.A.E. spline size for direct-drive engines and No. 40 for the geared engines.

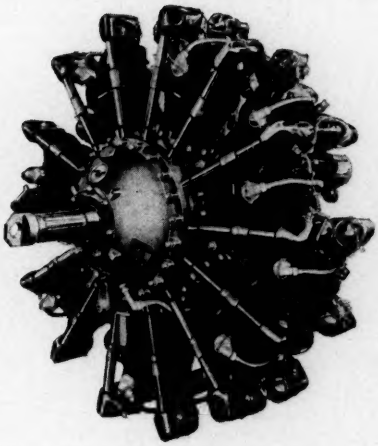
Eight mounting lugs are cast on the blower section and are arranged in four pairs so as to meet the mounting ring at points of attachment where the Vee braces in the conventional engine mount usually converge.

The accessory section is similar to Wasp and Hornet models and carries two Scintilla fourteen-cylinder shielded magnetos, the oil pumps, fuel pump, Stromberg carburetor, Pratt & Whitney combination hot spot and oil temperature regulator, provision for driving two gun synchronizers, two tachometers, and generator. The oil strainer and oil pressure relief valve are also mounted on the accessory section, while provision is made for oil temperature thermometer connections. A standard mounting flange is provided for the attachment of various types of starters.

The supercharger drive is similar to that used on the latest Pratt & Whitney Wasp and Hornet highly supercharged engines, but has increased clutch capacity.

In the geared engines, the Pratt & Whitney patented 3:2 planetary reduction gear is used. The distinctive feature of this gear is the means for equalizing the load on six pinions. It is a concentric gear of compact design and the application of it adds approximately eighty pounds to the weight of the direct-drive.

The direct-drive engines tested and delivered to date have been rated at 625 horsepower at 2100 revolutions per minute and the bare weight of the engine is approximately 1.33 pounds per horsepower.



TWIN WASP JUNIOR ENGINE.

Because of special features of design and light weight of the reciprocating parts, it is possible to operate safely at higher revolutions per minute than other radial engines of similar displacement. This makes possible ratings for the geared engines higher than those of the direct-drive.

Supercharging can be increased over that used to date, where the class of service may warrant.

The center of gravity of the engine is much closer to the engine mount than would be expected on an engine of this type. The small over-all diameter of 43⅞ inches permits of excellent vision in single engine tractor installations.

Both geared and direct-drive engines of the R-1535 Twin Wasp Junior model have been thoroughly block tested and both have been flown for a considerable number of hours. Engines for experimental service tests are being delivered to the Navy Department. Coincident with the development of the Twin Wasp Junior, other Pratt & Whitney models of the two-row radial type are being developed and tested.

A few of the more important general specifications follow:
Twin Wasp Junior-R-1535-Direct Drive.
Type: Air-cooled, Fixed two-row radial.
Number of Cylinders: 14 (two banks of seven).
Bore: 5 3/16 inches.
Stroke: 5 3/16 inches.
Displacement: 1535 cubic inches.
Compression Ratio: 6:1.
Blower Ratio: 10:1.
Rated Power: 625 horsepower at 2100 revolutions per minute.
Weight: 830 pounds bare (1.33 pounds per horsepower).
Length Over-All: 48¼ inches.
Diameter Over-All: 43⅞ inches.

BOILER PRESSURE AND TEMPERATURE PROGRESS.

Except in a few special cases, 1400 pounds per square inch represents the highest boiler pressure used in steam-raising today. One exception is the Langebrugge Station, where a Benson type boiler is installed, and at the Witkovitz Iron and Steel Works, Czecho-Slovakia, the boiler pressure, formerly 1700 pounds has been raised to 1800 pounds. At the Ford Motor Company's factory on the Thames the boilers are designed for 1400 pounds, the operating pressure being 1250 pounds; but generally modern pressures range from 400 to 600 pounds, the number of boilers working at higher pressures being very few. The Löffler boilers recently ordered for Moscow, however, are to be constructed for a pressure of 1900 pounds, the steam temperature being 932 degrees F. The feed water will be heated to 410 degrees F. by means of steam heaters fed by steam from the main plant and then further heated to 480 degrees F. by gases before being fed to the boilers.

Less difficulty appears to have been met in increasing steam pressures, even to 1400 pounds, than in raising the temperature. The changes in design necessitated by high pressures have been less than anticipated, though the riveted drum has had to give place to the solid forged drum for over 650 pounds. Welded drums are not yet considered acceptable in this country, but the attention which has been given during the last few years to the technique of welding, by boiler makers, may result in establishing this cheaper form of construction. High-pressure installations, on the whole, have proved satisfac-

tory, though the three standard pressures of 200, 300 and 600 pounds per square inch recommended in this country will probably continue for some time to dominate our practice, with 850 pounds as the upper limit where the requirements of process work demand a higher figure.

With regard to steam temperatures, 932 degrees F. at Langebrugge is at present the highest in Europe. There has, however, been an upward trend and the generally accepted limit is now 750 degrees to 800 degrees F. On the Continent over 800 degrees F. appears to be coming into wider use. A set recently supplied to Detroit (U. S. A.) by a British firm was designed to operate with steam at 1000 degrees F., and though that temperature has not yet been used, 957 degrees has been reached.

Improvements in turbine design and materials have overcome the earlier difficulties with high temperatures. Definite knowledge is accumulating steadily regarding the maximum stresses permissible with materials under various operating conditions and accordingly operation at higher temperatures is taking its place as standard practice.—"The Fuel Economist," April, 1932.

COLLOIDAL FUELS FOR MARINE SERVICE.

On the return of the Cunard steamship *Scythia* to Liverpool on Tuesday last, July 5th, after a successful round trip to New York, it was learned that the colloidal fuel, which was used experimentally in four of the furnaces of a single-ended boiler, was burned without any hitch. The results obtained seem to bear out the claims made for the efficiency and adaptability of the new fuel. It consists, it may be recalled, in a mixture of 60 per cent of crude oil with 40 per cent of pulverized coal by weight, so intermixed as to create a solution which will remain stable for some months. The tests made by Mr. A. W. Perrins, of the Cunard Steamship Company, and Mr. A. M. Wood, of the Wallsend Slipway and Engineering Company, Ltd., who accompanied the ship, seem to confirm this stability, as no trouble was experienced with the pumps, strainers, heaters, or burners, and the emptied fuel tanks were found to be quite clean. About 17 tons of fuel per day were burned, and there was, we understand, no undue smoke, while the only deposit was a free brown ash taken from the smoke boxes. The burners were cleaned a little more often than is necessary with ordinary boiler oil, but there was no tendency to slagging. The experiments also indicated the possibility of changing over quickly from boiler oil to oil-coal fuel, or *vice versa*, without incurring any extra expense. The steaming appeared to be quite good and somewhat better than with ordinary oil fuel. The specific gravity of the fuel, which remained constant throughout the trials, has been given as 1.1, compared with about 0.96 for ordinary boiler oil, which means that it will not float when discharged into the sea, while in case of leakage or fire, it can be easily flooded with water.—"The Engineer," July 8, 1932.

CENTRIFUGAL AIR EXTRACTORS.

The name of the late Professor Rateau will always be associated with the development of the rotary pump for handling gases, he having been responsible for the introduction of the rotary supercharger for use in conjunction with aeroplane engines in order to overcome the effects of loss of power when flying at high altitudes. The firm with which his name is associated have now produced a form of air extractor, the design of which is based upon the principles first employed by Prof. Rateau and which, it is claimed,

is capable of yielding a compression ratio of about 100, the suction pressure being 0.01 kilogram per square centimeter (0.142 pound per square inch). and the discharge pressure 1.05 kilograms per square centimeter (14.92 pounds per square inch), all the pressures being measured on the absolute scale. This centrifugal air extractor has been developed primarily for use in conjunction with a process for obtaining power from the heat energy of the sea at different depths in which process the extraction of air is of prime importance, but it is apparent that the pump can be employed for other purposes, notably for maintaining a high vacuum in a steam condenser working in conjunction with a turbine. For this purpose the makers claim that it is three times more efficient than existing types of air extractors. The apparatus is now being demonstrated in Paris.—"Shipbuilding and Shipping Record," June 2, 1932.

NEW HELICAL SPRING FORMULAS AND TABLES.

By A. M. Wahl.*

Recent research on the question of stress in helical springs axially loaded has shown conclusively that the ordinary helical spring formulas may give results greatly in error. This is particularly true for springs of small index, that is, for those having a small ratio between the coil diameter and the wire diameter. The error may amount to as much as 60 per cent.

Since ordinary spring tables are based on the accepted formulas, it follows that the tables are also in error. For this reason, new tables have been prepared, as presented here, based on more accurate formulas, which have been checked up by the research. The new formulas are given in the following:

$$S = K \frac{16\,Pr}{\pi d^3}$$

$$K = \frac{4c-1}{4c-4} + \frac{0.615}{c}$$

$$F = \frac{64\,Pr^3}{d^4 G}$$

in which

S = maximum shearing stress in spring;

P = load on spring corresponding to stress S;

r = mean radius of coil;

d = diameter of wire;

$$c = \frac{2r}{d} = \text{the ratio } \frac{\text{mean coil diameter}}{\text{wire diameter}};$$

G = modulus of rigidity of the material, or modulus of elasticity in shear; and

F = deflection per turn at load P.

In these tables, the modulus of rigidity G was taken as 11,500,000 pounds per square inch.

These formulas are the same as those ordinarily used for computing spring tables, except that the stress formula is multiplied by a factor K greater than unity, this factor being dependent on the ratio of the mean coil diameter to the wire diameter.

* Westinghouse Electric & Mfg. Co.

The tables give the load and deflection on springs of different sizes at a maximum stress of 100,000 pounds per square inch. To find the load or deflection at any other working stress, the values given by the tables should be multiplied by the ratio $\frac{\text{working stress}}{100{,}000}$. The tables give the wire sizes used by the company with which the writer is associated. It should be noted that the stress of 100,000 pounds per square inch, on which these tables are based, is *not* the recommended working stress, but is used merely for convenience.

The matter of choosing the working stress for various applications of helical springs is very involved and cannot be treated here in detail. The proper working stress depends on such factors as the kind of material, number of repetitions of stress, seriousness of failure, range of stress, possibility of corrosion, heat-treatment, etc. Also, in many cases, the possibility of vibration or surging of the springs must be considered.

The object of publishing these tables is to give designers a convenient and accurate means of determining maximum stress in helical springs without the necessity of laborious computation.—"Machinery," June, 1932.

THE ENGINEERING SILVER SOLDERS.

By Ernest A. Smith, A.R.S.M.

The increasing use that is being made of silver solders in the engineering industries has brought into prominence the importance of the alloys of the ternary system silver-copper-zinc, which form the basis of practically all the silver solders used in brazing operations. Although these alloys have long been used as solders for silversmiths' work, it is only since their extension to engineering practice that scientific investigators have given attention to their constitution and physical and mechanical properties. This increased interest in the system is undoubtedly due to the prominence which the alloys are now taking in all brazing operations of both ferrous and non-ferrous materials. These materials cover a very wide range, and in consequence the composition of the silver solders varies considerably according to the class of metal or alloy on which they are to be used. Recent research has done much to establish the constitution and structure of the silver-copper-zinc system, and some data have been furnished on the mechanical properties. These data are a valuable addition to the somewhat meagre literature on the subject, but further investigation is much needed, especially in relation to the strength of the brazed joints on different materials, and on the effect of differences in technique in the brazing operation itself. A brief review of the results of the most recent research on these ternary alloys, therefore, will not be without interest.

It has long been known that the addition of silver, even in comparatively small quantities, confers special properties on the ordinary brazing brasses and makes them more satisfactory to use; it also imparts increased strength at relatively high temperatures and greater resistance to corrosive influences in comparison with ordinary brazing alloys. Engineering silver solders vary considerably in regard to silver content, the range being from about 9 per cent or 10 per cent of silver in the lower grades, and from about 30 per cent to 70 per cent, or more, in the medium and higher grades. The higher grades may be considered as silver-copper alloys to which zinc, from small quantities up to about 20 per cent, is added to lower the melting point, and the lower grades as ordinary brazing brasses to which a small quantity of

Table 1. Maximum Loads, in P
from

Diameter of Wire, Inch		1/8	5/32	3/16
0.014	P	0.317	0.689	0.552
	F	0.0195	0.0237	0.0508
0.020	P	2.305	1.890	1.595
	F	0.0117	0.0203	0.0323
0.022	P	3.030	2.429	2.116
	F	0.0100	0.0180	0.0280
0.026	P	4.868	4.05	3.432
	F	0.0073	0.0137	0.0225
0.0286	P	6.35	5.31	4.522
	F	0.0059	0.0116	0.0190
0.0348	P	10.71	9.22	8.04
	F	0.0037	0.0080	0.0137
0.0410	P		14.47	12.62
	F		0.0055	0.0095
0.0475	P			19.00
	F			0.0072
0.0540	P			26.65
	F			0.0052
0.0625	P			
	F			
0.0720	P			
	F			
0.0800	P			
	F			
0.0915	P			
	F			
0.106	P			
	F			
0.121	P			
	F			
0.135	P			
	F			
0.148	P			
	F			
0.162	P			
	F			
0.177	P			
	F			
0.207	P			
	F			
0.235	P			
	F			

Note: P = Load at
squar
F = Deflecti
100,00
To find
stress

oads, in Pounds, and Deflections per Coil, in Inches, for Helical Springs from 1/8 to 7/8 Inch Outside Diameter

Outside Diameter of Spring, Inch									
	5/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4	7/8
0 37	0.563 0.0508	0.410 0.0950	0.324 0.1534						
0 08	1.595 0.0333	1.207 0.0646	0.9685 0.1077	0.7965 0.1574	0.6775 0.2194	0.592 0.2887			
9 80	2.116 0.0280	1.609 0.0572	1.294 0.0955	1.068 0.1409	0.908 0.1964	0.8065 0.2598			
37	3.433 0.0225	2.633 0.0457	2.134 0.0773	1.784 0.1167	1.511 0.1625	1.313 0.215	1.040 0.345		
16	4.532 0.0190	3.416 0.0394	2.822 0.0675	2.360 0.1024	2.035 0.1447	1.758 0.1922	1.390 0.3074		
80	8.04 0.0137	6.19 0.0297	5.05 0.0515	4.24 0.0799	3.567 0.1198	3.20 0.1541	2.535 0.2482	2.10 0.365	1.782 0.505
7 55	12.62 0.0098	9.86 0.0225	8.13 0.0405	6.87 0.0638	5.92 0.0930	5.22 0.1252	4.19 0.2066	3.45 0.303	2.935 0.423
	19.00 0.0073	15.15 0.0172	12.47 0.0320	10.62 0.0514	9.13 0.0750	8.10 0.1029	6.50 0.1716	5.40 0.258	4.60 0.360
	26.65 0.0053	21.33 0.0135	18.00 0.0256	15.41 0.0419	13.28 0.0622	11.79 0.0863	9.55 0.1453	8.01 0.2215	6.82 0.310
		32.35 0.0098	27.35 0.0197	23.5 0.0323	20.5 0.0495	18.13 0.0699	14.80 0.1203	12.35 0.1832	10.61 0.2615
		47.0 0.0070	40.5 0.0148	35.0 0.0255	30.8 0.0392	27.35 0.0567	22.35 0.0983	18.82 0.1523	16.20 0.219
			54.0 0.0115	47.0 0.0207	41.5 0.0325	37.0 0.0470	30.35 0.0841	25.5 0.1313	22.15 0.190
			76.7 0.0084	68.4 0.0155	60.7 0.0251	54.4 0.0371	44.9 0.0682	38.0 0.1036	32.9 0.1506
				101.2 0.0109	91.3 0.0185	82.2 0.0280	68.4 0.0531	58.1 0.0865	50.55 0.1232
				140.0 0.0074	129.6 0.0124	118.7 0.0211	99.5 0.0418	85.1 0.0694	74.2 0.1042
						159.4 0.0152	136.0 0.0336	117.2 0.0573	103.0 0.0874
						203.7 0.0129	175.0 0.0278	151.2 0.0484	133.1 0.0747
							223.5 0.0225	194.5 0.0404	172.7 0.0636
							281.3 0.0182	249 0.0324	220 0.0540
								379 0.0221	343 0.0389
								473 0.0157	427 0.0333

= Load at maximum stress of 100,000 pounds per square inch.

= Deflection per turn of spring at maximum stress of 100,000 pounds per square inch, or at load P.

To find load or deflection at permissible working stress, multiply the table values by the ratio

$$\frac{\text{working stress}}{100{,}000}$$

Table 2. Maximum Loa

Diameter of Wire, Inch		1	1 1/8
0.0410	P	2.65	
	F	0.557	
0.0475	P	4.01	3.53
	F	0.475	0.510
0.0540	P	5.93	5.24
	F	0.412	0.529
0.0625	P	8.25	8.15
	F	0.350	0 450
0.0720	P	14.19	12.5
	F	0.296	0 383
0.0800	P	19.32	17.32
	F	0.262	0.336
0.0915	P	28.9	25.6
	F	0.2164	0.2885
0.106	P	44.58	40.7
	F	0.1760	0.2262
0.121	P	66.0	62.9
	F	0.1465	0.1961
0.135	P	91.1	81.4
	F	0.1241	0.1669
0.148	P	118.1	106.2
	F	0.1068	0.1434
0.162	P	153.2	137.7
	F	0.0914	0.1282
0.177	P	198.2	178.2
	F	0.0791	0.1082
0.207	P	308	279
	F	0.0638	0.0827
0.225	P	387.5	364
	F	0.0496	0.0704
0.283	P	594	544
	F	0.035	0.0611
0.283	P		669
	F		0.0435
0.307	P		820
	F		0.0388
0.331	P		1007
	F		0.0295
0.362	P		
	F		
0.394	P		
	F		
0.430	P		
	F		
0.469	P		
	F		
1/2	P		
	F		
9/16	P		
	F		
5/8	P		
	F		

mum Loads, in Pounds, and Deflections per Coil, in Inches, for Helical Springs from 1 to 2 1/2 Inches Outside Diameter

Outside Diameter of Spring, Inches

1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/4	2 1/2
			Note. P = Load at maximum stress of 100,000 pounds per square inch.						
			F = Deflection per turn of spring at maximum stress of 100,000 pounds per square inch, or at load P.						
			To find load or deflection at permissible working stress, multiply the table values by the ratio $\frac{\text{working stress}}{100{,}000}$						
2.53 0.010									
5.34 0.539	4.89 0.660								
8.15 0.460	7.39 0.562	6.81 0.686	6.01 0.836						
13.5 0.332	11.25 0.4785	10.2 0.536	9.36 0.705						
17.33 0.338	15.5 0.435	13.97 0.5385	13.23 0.6385	11.77 0.744					
25.5 0.2885	22.35 0.360	21.1 0.4475	19.27 0.540	17.72 0.6385					
40.7 0.2983	34.5 0.360	33.9 0.3735	30.6 0.4545	27.2 0.540	25.5 0.638				
58.9 0.1943	52.3 0.251	48.55 0.3145	44.55 0.3886	41.9 0.460	38.35 0.544	35.5 0.638			
81.4 0.1669	72.5 0.2168	67.2 0.2700	61.95 0.3337	57.2 0.402	53.3 0.474	49.6 0.554	46.7 0.638	41.1 0.822	
106.5 0.1454	94.5 0.1909	88.5 0.2381	81.1 0.2944	75.2 0.3545	70.0 0.420	65.6 0.494	61.45 0.580	53.1 0.749	
137.7 0.1282	125.3 0.1664	115.9 0.211	105.0 0.258	98.3 0.3145	91.45 0.3727	85.5 0.440	80.5 0.508	71.5 0.6955	64.4 0.850
173.3 0.1088	162.3 0.1429	149.4 0.1836	137.0 0.2273	127.7 0.2781	119.5 0.3368	111.5 0.3968	105 0.4544	93.2 0.5945	84.3 0.7535
279 0.0627	256 0.1114	235.5 0.1432	216.5 0.1797	202.5 0.2208	188 0.2645	176.5 0.3145	166.7 0.3667	149 0.4836	133.9 0.619
364 0.0706	333 0.0864	295.5 0.1244	276 0.1562	256.5 0.1932	241.5 0.2365	225 0.2772	212.5 0.325	190.1 0.434	171.2 0.546
544 0.0511	500 0.0709	464.5 0.0943	431 0.1191	402 0.1493	377 0.1819	354 0.2187	333 0.2562	300 0.3453	273 0.448
668 0.0426	619 0.0611	571 0.0814	532.5 0.1062	496 0.1320	466 0.1602	439.5 0.1938	414 0.229	373 0.3128	339.6 0.404
930 0.0383	773 0.0511	719 0.0691	671 0.090	629 0.1138	590 0.1402	556 0.1698	525 0.2004	473 0.276	432.5 0.3605
1007 0.0295	945 0.0422	883 0.0586	829 0.0774	778.5 0.0986	733 0.1219	693 0.1481	662 0.1767	583.5 0.243	536 0.3165
	1232 0.0340	1153 0.0476	1068 0.0638	1019 0.0814	976 0.1027	914 0.1265	864.5 0.1512	761 0.2093	712 0.276
			1324 0.0534	1262 0.0688	1192 0.0864	1127 0.1062	1072 0.1291	970 0.1808	882.5 0.2408
				1590 0.0559	1515 0.0717	1452 0.0903	1368 0.1090	1243 0.1548	1138 0.2035
					1837 0.0609	1722 0.0762	1646 0.0943	1505 0.1352	1382 0.1787
							2084 0.0738	1883 0.1148	1745 0.1571
							2905 0.0536	2630 0.0834	2430 0.1227
								3470 0.0634	3230 0.0951

Table 3. Maximum L

Diameter of Wire, Inches			
		2 3/4	3
0.177	P	76.0	
	F	0.529	
0.207	P	122.4	112
	F	0.786	0.930
0.225	P	154.2	144
	F	0.890	0.944
0.263	P	250	229.2
	F	0.5635	0.667
0.283	P	303.2	285.5
	F	0.5085	0.635
0.307	P	392.5	360.2
	F	0.454	0.555
0.331	P	491	453
	F	0.4045	0.502
0.362	P	655	604.5
	F	0.3545	0.441
0.394	P	812	752
	F	0.309	0.388
0.450	P	1043	970
	F	0.2966	0.238
0.480	P	1269	1179
	F	0.2403	0.302
1/2	P	1613	1500
	F	0.208	0.262
9/16	P	2250	2095
	F	0.1644	0.218
5/8	P	3000	2805
	F	0.1340	0.173
11/16	P	3916	3722
	F	0.1081	0.144
3/4	P		4660
	F		0.117
13/16	P		
	F		
7/8	P		
	F		
15/16	P		
	F		
1	P		
	F		
1 1/8	P		
	F		
1 1/4	P		
	F		
1 3/8	P		
	F		
1 1/2	P		
	F		

laximum Loads, in Pounds, and Deflections per Coil, in Inches, for Helical Springs from 2 3/4 to 7 1/2 Inches Outside Diameter

Outside Diameter of Spring, Inches										
3/4	3	3 1/2	4	4 1/2	5	5 1/2	6	6 1/2	7	7 1/2
.0 129										
2.4 796	113 0.990	95.5 1.298		Note: P = Load at maximum stress of 100,000 pounds per square inch.						
6.2 990	144 0.844	122.5 1.151		F = Deflection per turn of spring at maximum stress of 100,000 pounds per square inch, or at load P.						
0 5635	229.2 0.667	197.3 0.901	171.9 1.217	To find load or deflection at permissible working stress, multiply the table values by the ratio $\frac{\text{working stress}}{100{,}000}$						
3.2 5085	285.5 0.525	245.6 0.721	214.5 1.009							
3.5 154	360.2 0.5535	313.5 0.802	275 1.089	243.5 1.415						
1 1045	453 0.502	390.5 0.755	342.2 0.993	304.5 1.298						
5 2545	604.5 0.441	524 0.6405	459 0.879	409 1.150						
8 809	752 0.388	652 0.569	576 0.7555	514 1.032	464 1.322					
943 4068	970 0.338	844 0.500	743 0.695	664.5 0.9215	602.5 1.177					
989 2403	1179 0.3022	1026 0.452	907 0.631	805 0.834	724 1.077	671 1.342				
512 308	1500 0.262	1302 0.3955	1160 0.560	1041 0.748	941 0.962	857 1.205	788 1.471			
350 1644	2095 0.2135	1836 0.325	1632 0.464	1472 0.628	1328 0.814	1214 1.026	1118 1.258	1033 1.518		
900 1240	2505 0.1726	2490 0.2726	2202 0.3915	1992 0.536	1811 0.700	1659 0.8825	1521 1.088	1417 1.320		
916 1021	3722 0.1441	3263 0.2272	2915 0.3205	2635 0.456	2398 0.600	2198 0.767	2030 0.950	1875 1.157		
	4660 0.1174	4140 0.1910	3750 0.2682	3390 0.3945	3054 0.519	2826 0.672	2615 0.836	2424 1.026	2258 1.232	
		5130 0.162	4660 0.2436	4250 0.3415	3860 0.458	3557 0.589	3275 0.736	3055 0.909	2823 1.093	
		6320 0.1368	5720 0.2008	5240 0.300	4810 0.403	4420 0.522	4085 0.653	3805 0.808	3535 0.972	
			6945 0.1511	6380 0.203	5840 0.2555	5360 0.464	4970 0.586	4640 0.725	4325 0.878	
			8280 0.1267	7590 0.229	7000 0.2125	6460 0.4125	6000 0.525	5590 0.650	5215 0.794	
				10470 0.1762	9710 0.248	9000 0.3210	8410 0.405	7810 0.521	7340 0.650	
					12980 0.1961	12090 0.2672	11230 0.3467	10560 0.441	9940 0.544	9330 0.669
						15650 0.2156	14610 0.2848	13810 0.3622	12980 0.453	12280 0.556
							18670 0.2353	17570 0.302	16430 0.3845	15700 0.4635

silver has been added. Silver solders are classed as hard solders, the term hard having reference to their relative fusibility, which varies from about 670 degrees to 850 degrees C.

Before dealing with the constitution of the ternary system silver-copper-zinc it is desirable very briefly to refer to each of the binary systems involved, viz., silver-copper, silver-zinc, and copper-zinc. All the alloys of the silver-copper system are malleable and ductile, even including the eutectic when suitably heat-treated. The alloys are moderately hard and are good conductors of electricity. The silver-zinc alloys are more resistant to tarnish than the silver-copper alloys of corresponding percentage composition, a fact first recorded by Peligot in 1864. The corrosion-resisting properties of the ternary silver-copper-zinc alloys of high silver content are largely due to the silver and zinc present. The silver-zinc alloys with less than 20 per cent of zinc are malleable and ductile, but those with from 20 per cent to 30 per cent are brittle. With from about 48 per cent to 50 per cent of zinc the alloys have a conchoidal fracture with a brilliant glassy surface and are very brittle. The hardness of the alloys increases with the percentage of zinc.

The constitution of the copper-zinc alloys is too well known to require detailed description here beyond pointing out that, in general, the three types of solid solution found in brasses differ considerably in mechanical properties. The α solid solution, up to a content of 30 per cent of zinc, is not particularly strong, but is exceedingly ductile. The β brasses, containing approximately from 36 to 50 per cent of zinc, on the other hand, are much stronger than the α, and are moderately ductile, and tough. The γ solid solution, containing more than about 50 per cent of zinc, is brittle and weak. The characteristics of the different solid solutions are important, in view of the fact that silver solders are essentially brasses of various compositions to which more or less silver is added, and the ease with which these solders may be manufactured is largely dependent on the proportions of copper and zinc present; silver may be regarded as taking the place of part of the copper. It is to be expected that the variations in the proportion of zinc in the brasses, and still more, the changes of constitution from α to β, &c., will influence profoundly the mechanical and physical properties of the alloys used as solders.

The liquidus diagram of this ternary system of silver-copper-zinc alloys was worked out in the United States by Handy and Harman, in 1929, but a more comprehensive examination of the system was made in the same year by Ueno.*

Ueno determined both the solidus and the liquidus, not only of the ternary silver-copper-zinc alloys, but also those of a number of alloys of the three binary systems concerned, and suggested a slight modification of the silver-zinc diagram. The liquidus, or temperature at which the alloy is completely fluid, is the one of most importance from the brazing point of view, and is frequently referred to as the "flow point," being the point at which the melted solder will flow freely on the surface of the material when properly protected from oxidizing influences. The flowing depends on surface tension, and this is dependent on the composition and constitution of the solder. The flow points of the binary silver-copper alloys are lowered by the addition of zinc on the one hand, and those of the binary copper-zinc alloys lowered by the addition of silver on the other hand. By varying the composition of the alloys, therefore, it is possible to obtain an extensive series of solders having a comparatively wide range of flow points, varying, as previously stated, from about 670 degrees C. to 850 degrees C.

* Mem. Coll. of Science, Kyoto Imp. Univ., vol. xii. Series A, page 347 (1929).

TABLE I.—*Melting Points of Silver-Copper-Zinc Alloys.*

Composition. Weight per Cent.			Temperature. Degrees Centigrade.		
Silver.	Copper.	Zinc.	Solidus.	Liquidus.	Plastic Range.
(i) *High Grade.—Silver, 50 per Cent. to 80 per Cent.*					
80	20	—	776	807	31
80	15	5	728	797	69
72	28	(Eutectic)	—	778	—
70	30	—	777	785	18
70	25	5	740	760	20
70	20	10	714	740	26
60	40	—	776	815	39
60	30	10	725	735	10
60	20	20	680	681	1
50	50	—	776	851	75
50	40	10	727	795	68
50	30	20	688	742	54
50	25	25	673	703	30
(ii) *Medium Grade.—Silver, 20 per Cent. to 50 per Cent.*					
40	60	—	776	887	11
40	50	10	728	845	117
40	40	20	684	790	106
40	32	28	—	735	—
40	30	30	722	725	3
30	70	—	776	934	158
30	60	10	734	999	265
30	50	20	689	844	155
30	42	28	673	785	112
30	40	30	—	780	—
20	80	—	777	970	193
20	70	10	735	948	213
20	60	20	693	900	207
20	50	30	674	835	161
(iii) *Low Grade.—Silver, less than 20 per Cent.*					
10	60	30	676	896	220
10	55	35	—	860	—
10	50	40	842	849	7
10	45	45	827	837	10
7	83	10	733	1,013	280
7	73	20	690	966	276
5	65	30	680	920	240
5	60	35	—	885	—
5	55	40	862	870	8
5	50	45	852	860	8
5	45	50	833	845	12

TABLE II.—*Silver Solder Specifications.*

Grade.	Composition. Weight per Cent.			Temperature. Degrees Centigrade.	
	Silver.	Copper.	Zinc.	Solidus.	Liquidus.
(i) *British Standard Specification* (*British Standards Institution*).					
A ..	60–62	27·5–29·5	9·0–11·0	—	710*
B ..	42–44	36·0–38·0	18·5–20·5	—	700
(ii) *American Specifications.*					
1 ..	10	52	38 (*a*)	820	870
2 ..	20	45	35 (*a*)	775	815
3 ..	20	45	30 (Cd, 5)	775	815
4 ..	45	30	25	675	745
5 ..	50	34	16	695	775
6 ..	65	20	15	695	720
7 ..	70	20	10	725	755
8 ..	80	16	4	740	795

* Approximate determinations by the author.

Note.—(*a*) The addition, to Grades 1 and 2, of not more than 0·5 per cent. of cadmium, to assist in manufacture, shall not be considered a harmful impurity.

Grades 1, 2, and 3 are suitable for brazing purposes which require a solder that flows more readily than the ordinary copper-zinc brazing solders. Grades 4 and 5 flow freely at a still lower temperature, and are recommended for use where strong joints are required and in such cases where heating to a sufficiently high temperature to flow Grades 1, 2, and 3 would be injurious to the article being soldered.

Grades 6, 7, and 8 are higher grade silver solders and are recommended for special cases where a high degree of malleability and ductility is required. The solidus and liquidus temperatures are those which accompany the specifications.

Grade.	Composition. Weight per Cent.			Melting Point.
	Silver.	Copper.	Zinc.	Deg. Cent.
(iii) *U.S. Navy Specification* (47 *S.* 13*a*).				
0	20–25	45–50	Remainder	—
1	44–50	35–45	Do.	—
2	64–70	18–23	Do.	—
(iv) *German Specifications.*				
Silver Solder in Grain form :—				
4	4	50	46	740
9	9	43	48	720
12	12	36	52	690
Silver Solders in Strips :—				
8	8	50	42	670
25	25	40	35	620
45	45	30	25	600

Example of application of these solders:—Jointing of brass containing 58 per cent. copper and over, and also for copper and bronze pieces.

A very good indication of the flow points of ternary alloys of silver-copper-zinc, differing widely in composition, is afforded by the figures in Table I, compiled from Ueno's results. The figures also show the duration of the plastic stage, which varies considerably for the different alloys, and is mainly influenced by the zinc content. This phase represents, of course, a mixture which is partly liquid and partly solid.

The silver solders in use in engineering practice are well exemplified in the specfications published by the British Standards Institution in this country, and by similar bodies in the United States and in Germany. These typical specifications are given in Table II.

The issue of these specifications, prepared by engineers and metallurgists, in collaboration, not only emphasizes the growing importance of silver solders for engineering purposes, but also gives an indication of the compositions of the alloys that have been found to be most generally useful for brazing and soldering purposes on all classes of material. In addition to these specifications, most of the leading electrical concerns have other special specifications drawn up to suit their own particular requirements. With regard to the alloys containing a small percentage of cadmium, included in the American specifications, the addition of this metal, which melts at 320 degrees C.—that is, 100 degrees below the melting point of zinc—lowers the melting point of the solders in some cases, but its chief function appears to be to facilitate the preparation of the solders, since research by Jenkins,* has shown that the presence of cadmium in small quantity, in zinc, improves the rolling qualities, and increases the strength of the latter metal. The effect on the fusibility of solders of substituting cadmium for zinc depends on their composition, and has been studied by Leach,† who concluded that the flow point is slightly raised, in the case of alloys containing 65 per cent or more of silver, whilst in the alloys containing relatively low percentages of silver the substitution of from 4 to 5 per cent of cadmium lowers the flow point. Malleable alloys of medium silver content can be produced with a higher percentage of zinc and cadmium combined than with zinc alone, thus giving slightly lower flow points.

The addition of nickel to silver solders has been advocated, but very few data are available on its advantages or otherwise. Leroux and Raub‡ have recently investigated the effect of small quantities of nickel on the properties of the ternary silver-copper-zinc alloys, and have pointed out that the tendency to coarse grain growth which results from the annealing of these alloys can be prevented by small additions of nickel (0.3 to 1.5 per cent). The solid solubility of nickel in the α silver-copper-zinc phase is less than 0.1 per cent at room temperature, and 0.2 to 0.3 per cent at 700 degrees C. They find also that the addition of nickel produces a relatively large increase in the tensile strength, but a reduction in the elongation; it also increases the hardness, but has comparatively little effect on the flow point. These conclusions are based on experiments on a silver-rich ternary alloy containing 83.5 per cent silver, 12.3 per cent zinc, and 4.2 per cent copper. Further work on the effect of nickel on ternary alloys of lower silver content will be looked for with interest, in view of the increase of strength in brazed joints, which apparently results from its presence. Where the color of the brazed joint is an important consideration, high-grade solders are used, as alloys containing over 60 per cent of silver are silver-white, whilst those with from 60 to 45 per cent of silver have a slight yellow tint, and below 45 per cent silver the alloys become increasingly yellow until the characteristic yellow of brass is reached. Moreover, the whitness is retained over a longer range in high-grade and medium grade alloys when zinc is also present.

* Engineering, vol. cxxii, page 614 (1926).
† Proc. Am. Soc. Test. Mat., vol. xxx, page 493 (1930).
‡ Z. Metallkunde, vol. xxiii, page 58 (1930).

TABLE III.

Grade of Solder.	Silver.	Zinc.
	Per cent.	Per cent.
High grade	50 to 80	0 to 18
Medium grade ..	20 „ 50	18 „ 30
Low grade or common	5 „ 20	30 „ 50

TABLE IV.—(*Figures derived from Ueno.*)

Composition. Weight per Cent.		Temperature. Degrees Centigrade.	
Copper.	Zinc.	Solidus.	Liquidus.
90	10	1,020	1,040
67	33	913	930
50	50	862	868
34	66	800	827

TABLE V.—TENSILE STRENGTH, ELONGATION, AND ELECTRICAL CONDUCTIVITY OF SILVER SOLDERS.

Composition per Cent.			Tensile Strength. Tons per Square Inch.	Elongation on 2 In. Per Cent.	Electrical Conductivity. Copper 100 per Cent.	Conductivity Resistance. Microhms per Cubic Centimetre.
Silver.	Copper	Zinc.				
9	51·0	40·0	21·8	16·0	20·5	8·33
20	45·0	30·0 (Cd, 5)	23·7	12·0	24·4	7·69
25	52·5	22·5	—	—	24·4	7·69
30	38·0	32·0	—	—	24·4	7·69
40	36·0	24·0	26·0	6·2	19·7	8·65
45	30·0	25·0	—	—	19·0	8·97
46	38·5	15·5	22·3	9·0	—	—
47	29·0	24·0	26·1	16·0	—	—
50	28·0	22·0	—	—	19·7	8·65
50	32·5	17·5	24·8	9·0	—	—
50	34·0	16·0	—	—	24·1	7·04
60	25·0	15·0	28·7	7·7	20·5	8·33
65	20·0	15·0	29·0	34·0	21·3	8·01
70	30·0	—	25·7	25·0	—	—
70	25·0	5·0	22·4	9·5	—	—
70	20·0	10·0	—	—	26·7	6·41
72	28·0	—	—	—	77·1	2·24
75	20·0	5·0	—	—	38·1	4·48
75	22·0	3·0	18·6	5·3	53·4	3·20
80	16·0	4·0	22·3	16·0	45·8	3·73

Mention must also be made of the effect of zinc on the mechanical properties of the ternary alloys used as solders, and of the necessity for limiting its use in relation to the silver content. The mechanical properties of solder alloys are also greatly influenced by the presence of impurities, especially lead; hence it is of fundamental importance that pure metals should be used in making all such alloys. The zinc used should be of electrolytic quality, or good quality spelter (G.O.B.), or in the form of good quality brass of known composition. As the silver content of the solders is reduced, the flow point will be correspondingly raised unless an abnormal proportion of zinc is added, in which case brittle alloys will result. In cases in which a very hard solder is required a binary alloy of silver and copper is sometimes used. The eutectic containing 72 per cent silver, and 28 per cent copper, melts at about 778 degrees C., a temperature sufficiently low for some purposes, but it is more usual to add a little zinc to a binary alloy of higher silver content to get a solder of the desired melting point. Thus a good medium solder consists of silver 75, copper 20, and zinc 5 per cent, respectively. The permissible proportions of zinc for the different grades of silver solder, based on silver content, are approximately as shown in Table III. As an example, it is possible to produce a malleable and ductile solder containing 65 per cent silver, 20 per cent copper, and only 15 per cent zinc, which flows freely at approximately 700 degrees C. By increasing the zinc, however, the flow point may be lowered to about 680 degrees C., but with a falling off in ductility and strength, and a tendency to brittleness.

With reference to brazing solders containing a comparatively small amount of silver, the advantages to be derived from such addition appear to have been first pointed out by Schwirkus in 1894, who showed that brazing solders tested on different qualities of brass gave improved results when containing from 2 to 12 per cent of silver. A considerable number of copper-zinc alloys, which vary in composition according to the melting point required, are now employed industrially as brazing solders, for brazing brass and other copper alloys, and for nickel alloys, and iron. The most common mixture, known as "brass solder," contains copper and zinc in equal proportions, and is almost universally employed for hard soldering. In cases in which a brass solder with appreciably lower melting point is necessary, an alloy of one part copper, and two parts of zinc is generally employed. For brazing iron the solder most frequently employed contains two parts of copper and one part of zinc.

"Brazing solder for use in acetylene soldering usually contains between 88 per cent and 90 per cent of copper, the balance being zinc. The Admiralty standard brazing metal for this class of work contains 90 per cent copper and 10 per cent zinc."* The approximate melting points and flow points of these brazing alloys are shown in Table IV. The addition of silver lowers the melting point of these brazing solders still further, and improves their general properties. Thus 10 per cent of silver added to the 50/50 brass gives a flow point of 837 degrees C.; to the 67/33 brass, 896 degrees C.; and to the 90/10 brass, 999 degrees C.

With careful regulation of the composition, silver solders of all grades have comparatively high tensile strength, and are malleable and ductile, but greater care is needed in the preparation of alloys containing a relatively high proportion of zinc. All grades can be rolled into thin sheet or strip, and drawn into wire, or prepared in granular form, and are sufficiently ductile and tough to make strong joints which will withstand bending stresses, shocks and vibrations, even at elevated temperatures, to a greater extent than the ordinary brazing brasses.

* Engineering Non-Ferrous Metals and Alloys, Aitchison and Barclay.

A few ductility tests made some years ago by the writer, with the Erichsen machine, on silver-solder sheet, containing 79 per cent silver, and 66 per cent silver, respectively, in the annealed condition, gave depths of cup of 12 millimeters in the one case, and 11.3 millimeters in the other. The rolling and wire-drawing of medium and low-grade silver solders call for the same precautions as are necessary in the case of the brasses, especially in regard to the casting temperature and conditions, and to the temperature and atmosphere of the annealing operation.

Very few scientific data are available relative to the mechanical properties of the silver solders, the latest published in this connection being those already referred to by Leach, who has determined the tensile strength, elongation, and electrical conductivity of a series of some twenty alloys of different composition. The tensile strength of silver solders as cast will vary from 18 tons to 27 tons per square inch. The tensile strength and elongation of the alloys tested by Leach, and their compositions, are shown in Table V. The tests were made on cast bars having a cross section measuring 0.1-inch by 0.3-inch, and ranging in silver content from 9 per cent to 80 per cent. With reference to these figures, Leach remarks that "these bars were cast in iron moulds and the rate of chilling was probably much more rapid than when the solder is used. The elongation figures were very erratic and examination of the surfaces at the break showed that this was partly accounted for by too rapid chilling and consequent unsound castings."

The tensile strength is of value because it is a typical measure of the general strength of the solder, and there are applications where this and also impact strength are important. In most cases, however, soldered and brazed joints exist in practice in a state of shear stress, although but lightly stressed in many cases, and this stress has been taken as the criterion of mechanical strength in the research on jointing of metals now being conducted by the British Non-Ferrous Metals Research Association. The testing of ordinary single lap or double lap joints is attended by many difficulties, and so far no data have been published regarding test of joints made with silver solders of different composition, and on different classes of material. Such tests are greatly influenced by the composition and constitution of the solders used, the nature of the material on which the joint is made, and the technique of the soldering or brazing operation. A comprehensive investigation on the physical properties of silver solders and on the strength of soldered and brazed joints would be very welcome at the present time, and would doubtless lead to still further extension of their uses.

Silver solders are remarkably fluid when in the molten state, and penetrate interstices which other classes of solder would fail to fill. Intercrystalline penetration is very noticeable in some cases, especially with metals of large grain structure. Applications of silver solders in electrical machinery involve consideration of the electrical resistivity of the alloys used. In some cases where the number of joints is large the collective electrical resistance may be appreciable. The electrical conductivity of silver solders is largely controlled by their composition, and is more especially dependent upon the percentage of zinc present. Table V gives the electrical conductivity of a useful range of silver solders as determined by Leach. The superior corrosion-resistance of silver solders over the ordinary brazing brasses has been largely responsible for their increased use in recent years, but this valuable property is mainly confined to solders of high silver content. Satisfactory information on this important question can only be obtained by testing the solders under the conditions to which they will be exposed in actual practice. In general, silver solders are as resistant to corrosion as are the non-ferrous metals and alloys employed for their corrosion-resisting properties.

They are used extensively in the manufacture of chemical apparatus, and the silver-copper eutectic, which melts at 778 degrees C., can be used as a silver solder in those cases where the presence of zinc is harmful. They are attacked by nitric acid, and discolored by sulphur.

The grade of solder to be used for any specific purpose is dependent on many factors, the nature of the metal or alloy to be joined being one of the most important. Leach has summarized some of the more important industrial applications of silver solders which include the following: For brazing copper all grades of silver solder may be used, but there are many instances in which those containing the higher percentages of silver are to be preferred, as in the case of rod or sheet which is to be cold-worked to any extent, in which case 60 per cent to 80 per cent silver solders are advisable. Silver solders make strong ductile joints on all compositions of brass, and solders containing from 9 per cent to 10 per cent are largely used, but those containing 20 per cent of silver or more are preferable, especially when the relatively low melting point of the brass justifies the use of high-grade solders in order to prevent damage to the brass from overheating. All grades of nickel-brass (so-called nickel silver) which may contain from 5 per cent to 35 per cent of nickel, can readily be soldered with silver solders, and the lower grade solders can be used where color is not an important consideration. Good white joints are obtained when solders with over 50 per cent of silver are used. Thus, a solder frequently employed on good quality nickel silver, containing over 18 per cent of nickel, consists of silver, 67 per cent; copper, 23 per cent; and zinc, 10 per cent. An average solder for nickel-silver contains about 20 per cent silver.

Silver solders can be used to advantage on nickel itself, and on nickel alloys, such as Monel metal, and Corronil, and other alloys that approximate in composition to the Monel-metal type. For fastening Monel-metal turbine blades in position, solders containing about 60 per cent or 42 per cent of silver are in general use, such as those given in the British Standards Institution specifications A and B, shown in Table II; Grade B is used in Admiralty turbine work. Other solders recommended for Monel metal and similar alloys are, 70 per cent silver, 23 per cent copper, and 7 per cent zinc; 65 per cent silver, 25 per cent copper, and 10 per cent zinc; and 50 per cent silver, 35 per cent copper, and 15 per cent zinc. Aluminum bronzes, *i.e.*, alloys of copper and aluminum containing from 8 per cent to 10 per cent of aluminum, can be brazed with silver solders, but the film of oxide which forms interferes seriously with making good joints, and care is required in heating and fluxing. All grades of silver solders are used for brazing iron and steel and strong joints are obtained. Leach states that "The use of silver solders for joining steel band saws is a typical example where they are extensively used, because they melt at lower temperatures than base-metal brazing solders, and make strong joints which stand the heavy stresses to which band saws are subjected." They are particularly satisfactory for brazing iron and steel parts to non-ferrous metals and alloys having relatively low melting points. Attempts have been made to use silver solders for brazing stainless steels and iron, and Leach considers that silver solders containing 10 per cent or less of silver, and relatively high zinc content, give better results than solders that contain over 50 per cent silver. In a recent paper, however, Hatfield* states that the brazing and soldering of chrome-nickel steel joints is not very extensively employed, because the temperature necessary for these operations usually lies within the range which is unsuitable for the metal. The process of soldering, and of brazing, itself is largely based on empirical knowledge, and until quite recently the

* Engineering, vol. cxxviii, page 567 (1929).

chemical and physical aspects of the subject have received very little attention. However, as the result of recent investigations carried out by the British Non-Ferrous Research Association, and others, the process has been placed on a more scientific basis. The method of heating calls for careful consideration in view of the temperature required to melt silver solders. Satisfactory results are obtained with torches designed to use air and gas, oxygen and gas, or oxygen and acetylene. Electrical heating is also being adopted in many cases with very satisfactory results. For many years silver solders were regarded by engineers as an expensive luxury, and the price of silver in pre-war days perhaps somewhat justified this conclusion. The considerable expansion in their use, however, during the past few years, coupled with the abnormally low price of silver at the present time, has done much to change this view, and their advantages are now being appreciated and their use considered in industries where hitherto they have not found application. As the result of this expansion in the employment of silver solders there has arisen the necessity for some more definite information regarding their physical properties and characteristics, and it was with a view to supplying this information that the researches reviewed in this article were undertaken. Although these researches are a valuable addition to the scientific and technical literature of the subject, further research, as already pointed out, is a necessity if silver solders are to continue steadily to gain ground. A factor of importance in the use of silver solders, and one that is frequently overlooked, is that the scrap has a market value, owing to its silver content, and, if collected separately, and kept clean, can be sold. —"Engineering," April 15, 1932.

THE INFLUENCE OF DESIGN ON BRASS AND BRONZE CASTINGS.

By Lewis H. Fawcett.*

ABSTRACT.

The author stresses the attention which should be given the designing of castings in relation to foundry production. Selection of the alloy to be used is important, as is the determination of molding position to prevent shrinkage. Composition of some bronzes and brasses used by the Navy Department are given, together with the required physical properties. The fundamentals of design are discussed and many examples of poor design are shown with improvements which can be made to overcome the defective design. Evenness of metal section is stressed as of the greatest importance in securing sound castings.

1. Engineers responsible for the design of castings required in engineering projects give careful consideration to the composition of the alloy selected and its mechanical properties, with sufficient allowance for the proper factor of safety. However, in many instances this same consideration has not been given to the foundry problems involved in the production of the casting, and their influence on the ultimate value of the casting.

2. Unfortunately, reliable data on this subject are not always readily obtainable in concise form for the use of designers. It is the purpose of this paper to outline briefly some of these problems and to present some simple illustrations of good and poor design for brass and bronze castings.

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SELECTION OF ALLOY.

3. In the selection of an alloy, consideration should be given not only to meeting the requirements of the design but also to aid production and minimize foundry difficulties. When possible, an alloy should be selected permitting the use of a minimum amount of gates and shrink heads to obtain the maximum yield from the metal melted.

4. A brass with high shrinkage may have a low intrinsic value and yet have a high manufacturing cost, due to the excess metal needed for heads and gates and the additional molding necessary when using such an alloy. Such a brass would have a tendency to form scurf on its surface; but, in general, it would have a fairly uniform grain size so that the interior metal would about equal the surface metal in soundness.

5. A bronze would give a smooth, even, close-grained surface of uniform pleasing color and be produced with less molding labor and less excess metal in heads and gates. The bronze casting, however, probably would have a relatively coarse-grained interior structure as compared with the structure of the casting's exterior surface.

6. Table 1 gives the composition and mechanical properties of two well-known bronzes, and similar information regarding brass is shown in Table 2.

Bronze Cylindrical Bushing.

7. The cylindrical bushing shown in Figure 1, if made of bronze, in all probability would be cast in a vertical position, as shown in A of Figure 1. The shrink head would be placed on the top flange.

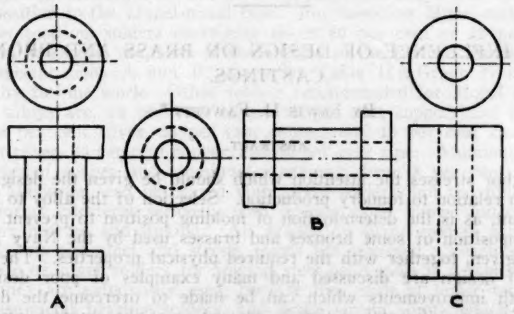


FIG. 1—SIMPLE BRONZE CYLINDRICAL BUSHING. *A*: PROBABLE CASTING POSITION, WITH SHRINK ON TOP OF FLANGE. SOME SHRINKAGE IS TO BE EXPECTED IN FLANGE. *B*: IF LARGE AMOUNT OF METAL IS REMOVED IN MACHINING, THIS WOULD BE PREFERRED POURING POSITION, AS COMPARED WITH *A*. IN POSITION *B*, HEADS WOULD BE LOCATED ON TOP OF BOTH FLANGES. *C*: PREFERRED DESIGN FOR BUSHING TO BE POURED OF A HIGH-SHRINKAGE BRASS OR MANGANESE BRONZE.

8. This, of course, does not give ideal conditions during solidification, as the metal must feed the bottom flange through the relatively thin cylinder. The thin section would solidify before the bottom flange and thus cut off the supply of liquid metal during the latter part of the freezing period of the bottom flange. Nevertheless, this procedure would result in a satisfactory

Table 1

Composition and Mechanical Properties of Bronze

Chemical Requirements

Navy Department Specifications.	Composition.	Cu, %	Sn, %	Zn, %	Pb, %	Fe, max. %	Ni, max. %	P, %	S, max. %	Sb, max. %	Other Elements, max. %
Phosphor Bronze 46-B-5f	Desired.	88.0	8.0	4.0	0	0	....	0	0	0	0
	Permissible.	85.0-89.0	7.5-11.0	1.5-4.5	0-1.0	0.25	0.75	0.5	0.05	0.25	0.35
Gun Metal 46-M-6e	Desired.	88.0	8.0	4.0	0	0	....	0	0	0	0
	Permissible.	86.0-89.0	7.5-11.0	1.5-4.5	0-0.30	0.10	0.75	0.05	0.05	0.25	0.15

Mechanical Requirements

Navy Department Specifications.	Tensile Strength in Lbs. per Sq. In., minimum	Elongation in 2 in., minimum
Phosphor Bronze 46-B-5f	35,000	18.0
Gun Metal 46-M-6e	40,000	20 0

Table 2

Composition and Mechanical Properties of Brass

Chemical Requirements

Navy Department Specifications.	Composition.	Cu, %	Zn, %	Pb, max. %	Al, max. %	Mn, max. %	Fe, max. %	Sn, max. %	Other Elements, max. %
Naval Brass 46-B-10d	Desired.	62.5	36.3	0	0.2	....	0	1 0	....
	Permissible.	60.0-65.0	†	0-1.0	0.5	....	0.5	0 5-1.5	....
Manganese Bronze 49-B-3c*		55.0-60.0	38.0-42.0	0.20	1.50	3.50	0.40-2.00	1.50	0 20

Mechanical Requirements

Navy Department Specifications.	Tensile Strength in Lbs. Sq. In., minimum.	Elongation in 2 in., minimum.
Naval Brass 46-B-10d	30,000	15.0
Manganese Bronze 49-B-3c*	65,000	20.0

* A high-test brass.
† Remainder.

casting, and if too much metal was not removed in machining, the casting would withstand a high hydrostatic pressure.

9. If, on the other hand, a large amount of the fine-grained surface metal was removed by machining, it probably would leak (under water pressure) at the junction of the side wall and bottom flange, due to the relatively coarse-grained interior metal allowing the water to seep through. By surface metal is meant the metal in contact with the sand; the interior and exterior of the cylinder would be alike in this respect.

10. To insure more perfect feeding the cylinder may be cast horizontally, as shown in B of Figure 1. In this case a head would be located on top of each flange. The thin body of the casting would solidify before the end flanges, but liquid metal in the head would be available for feeding the flanges during the whole time of their solidification.

11. If the casting was as simple as the one shown in the sketch, it could be cast in this manner. However, many castings which involve these principles have a much more complicated contour, and it is more practical to pour them in a vertical position.

Brass Cylindrical Bushing.

12. If the bushing was to be made of brass or manganese bronze (a high-shrinkage brass), it would be more satisfactory to pour a hollow cylinder as shown in C of Figure 1. This method would insure perfect feeding, as the casting is of uniform thickness throughout. Furthermore, it would allow the dross to rise and collect in the shrink head above without any tendency to be trapped in corners.

13. The cylinder then could be machined to the contour of the bushing. The machining would remove any scurf that might be present on the surface and would not affect the ability of the casting to withstand hydrostatic pressure because the metal does not have a tendency to form a relatively coarse-grain interior, as compared with bronze.

14. Considering all the factors involved, it probably would be more economical to make this bushing of bronze rather than of brass—especially so where the casting is required to withstand water pressure.

Considerations in Selecting Proper Alloy.

15. Figure 2 shows an exaggerated example of a casting that would be made more satisfactorily of brass than of bronze. The casting is 18 inches in diameter, 6 feet long and has a 3-inch diameter hole which is to be cored out with sand. The result is a large volume of metal surrounding a small

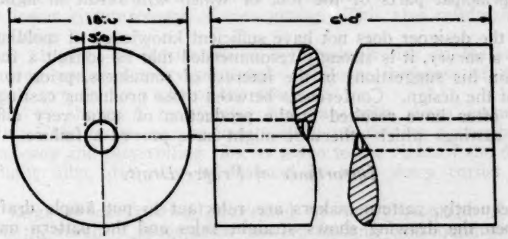


FIG. 2—EXAGGERATED EXAMPLE OF CASTING THAT WOULD BE MADE MORE SATISFACTORILY OF BRASS THAN OF BRONZE.

area of sand. The metal is, in turn, surrounded by the refractory sand of the mold which, of course, retards its rate of cooling.

16. If this job was made of bronze there would be a tendency for "tin sweat" to penetrate the sand core, making the removal of the core extremely difficult and resulting in a hard, irregular surface within the hole. Tin sweat is hard and difficult to machine. It frequently has a tin content of about 20 per cent, but may run higher or lower than this figure. Other things being equal, the higher the tin content of the bronze, the greater the tendency to form tin sweat.

DESIGN.

Uniformity of Cross-Sectional Area.

17. The ideal casting should be designed to have the same cross-sectional area throughout at the time of pouring. Of course, such uniform thickness is not to be expected in most castings but should be provided wherever practical. At times the allowance for machine finish may cause non-uniformity, although the final machined article may be uniform.

18. When uniform cross-sections are not possible, an abrupt change from one cross-section to another should be avoided by the liberal use of fillets, which will eliminate right-angle bends and sharp corners and greatly aid in the production of sound castings. Wherever sudden changes in section occur, the lighter section may have completed liquid shrinkage and solid contraction started, whereas the heavier section may still be in the liquid shrinking status, resulting in a rupture or strain at the union of the two sections if proper fillets or brackets are not employed at their intersection.

19. When it is necessary to attach lugs, bosses, brackets or other projections to the body of a casting, uniformity of metal should be maintained as far as practical, even though it requires the coring out of such projections. An intended solid boss, attached to the casting, may be weaker than a lighter boss that has been cored out but is entirely sound.

20. Uniform cross-section, the proper use of fillets, rounded corners, coring of heavy lugs and bosses, the avoidance of sharp corners and right angle bends are details of design that greatly aid in the manufacture of sound castings.

Cooperation of Designer and Foundryman Recommended.

21. Furthermore, in the interest of sound castings it is essential that the designer visualize the various molding methods applicable to his design so that he may make modifications which will permit the proper feeding of all the principal parts of the job, or which will result in lightening the casting.

22. If the designer does not have sufficient knowledge of molding practice for such a survey, it is strongly recommended that he consult a foundryman and obtain his suggestions in the interest of soundness, prior to the completion of the design. Conferences between those producing castings and the designer often have resulted in the production of some very difficult and intricate castings which otherwise might have proved a failure.

Importance of Proper Draft.

23. Frequently, pattern makers are reluctant to put ample draft on patterns when the drawing shows straight sides and the pattern maker does not know whether draft would affect the finished product. When possible, the drawing should specify a taper or authorized draft.

24. Sufficient draft will aid in the removal of the pattern from the sand without undue rapping, with the result that the sand will remain firm and thus obviate inaccuracy in the dimensions of the casting. Proper draft results in the casting being a truer replica of the pattern, decreases losses due to mold troubles, and is one of the largest factors in increasing production.

Examples of Good and Bad Design.

25. Accompanying sketches and photographs give some examples of good and bad design of brass and bronze castings.

26. Figure 3 shows the intersection of two members of equal thickness. The sharp corner shown in poor design (at the left of the drawing) should be eliminated by a gradual tapering from one section to the other, as shown in the good design to the right. At times the tapering may take the form of only a small fillet to avoid the right-angle bend, but in any event it will greatly aid in the elimination of stress concentration at such an intersection and is most essential at the junction of thick and thin sections.

27. In Figure 4 a too-liberal use of fillets at the intersection of ribs has caused an increase in the width of the intersection and thus has caused non-uniformity at this point. This may be overcome, as shown on the right of the drawing, by coring out the center of the intersection down to the body of the casting.

28. Figure 5 gives an example of staggered ribs designed to avoid intersections. The fillets also are slightly reduced.

29. The diagonal bracket shown as poor design in Figure 6 is a common error. The bracket has been made larger than the adjoining metal to add strength to the casting, but it has quite the opposite effect. The two sides of the triangle will solidify before the hypotenuse and thus enter the solid contracting stage with a tendency to shrink away from the thicker member before the latter is completely solid. The bracket should be of equal thickness whenever possible, as shown at the right of the drawing.

30. Figure 7 shows poor and good design for bolting plates or pads. Such projections should be cored out so that the cross-section of these members will be as uniform as possible with the body and thus avoid unnecessary increases in local thickness.

31. Figure 8 gives another illustration of coring out of a projection to maintain, as far as possible, equal thickness of metal. The lug is changed to a flange in this case.

32. Figure 9 shows a great reduction in metal thickness by modification of design without decreasing the bolting or bearing surface of the projection.

33. In Figure 10 uniform cross-section is obtained by coring out a section around the lower circumference of the casting. This also results in lightening the casting.

34. Figure 11 gives an illustration of a threaded collar on the end of a casting causing non-uniformity of metal. Uniform thickness of metal is obtained as shown on the right of the drawing.

35. Figure 12 shows a rectangular key-way with sharp corners and angular ribs. This design may be improved by substituting a rounded ridge over the key-way and easy-rolling ribs, as given to the right of the drawing. A very slight fillet also may be inserted in each sharp corner of the key-way.

36. The large bronze casting illustrated by the photograph shown as Figure 13 is the result of cooperation between designer and foundryman, and represents practical example of the principles previously mentioned. The casting has a large flange on each end and many projections on its body.

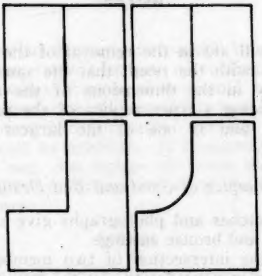


FIG. 3—POOR (LEFT) AND GOOD (RIGHT) DESIGN OF INTERSECTIONS.

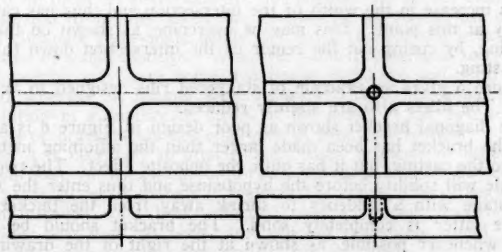


FIG. 4—*Left*: RIB INTERSECTION OF POOR DESIGN. *Right*: BETTER DESIGN OF RIB INTERSECTION.

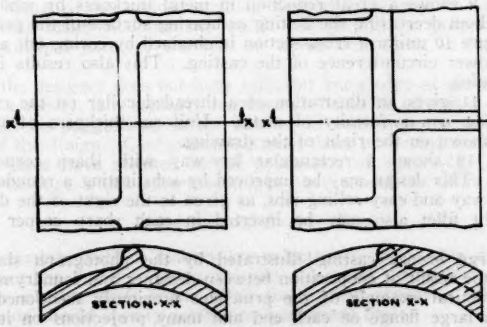


FIG. 5—EXAMPLE OF STAGGERED RIBS DESIGNED TO AVOID INTERSECTIONS AT RIGHT. AT LEFT, ORIGINAL DESIGN WHICH GIVES UNDULY HEAVY METAL THICKNESS AT INTERSECTIONS OF RIBS.

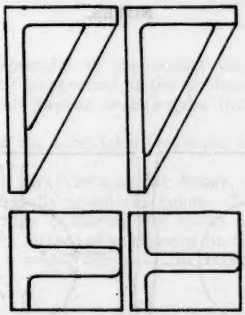


FIG. 6—*Left*: BRACKET SUPPORT IS TOO HEAVY, CAUSING WEAKNESS DUE TO UNEVEN METAL SECTION. *Right*: THIS FAULT CORRECTED.

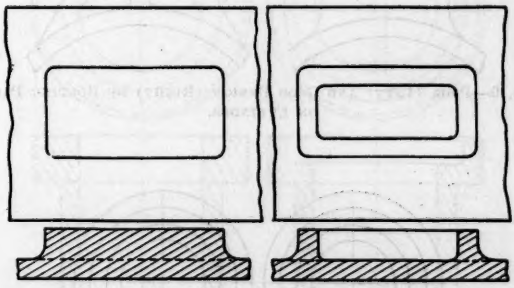


FIG. 7—POOR (LEFT) AND GOOD (RIGHT) DESIGN FOR BOLTING PLATES OR PADS.

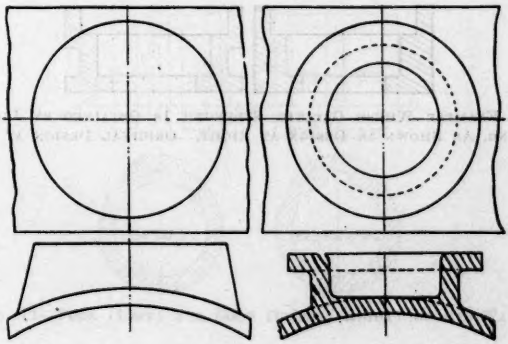


FIG. 8—POOR (LEFT) AND GOOD DESIGN (RIGHT) OF PROJECTION ON CYLINDER.

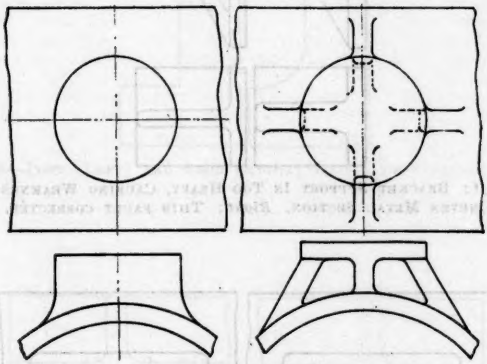


FIG. 9—POOR (LEFT) AND GOOD DESIGN (RIGHT) OF BOLTING PADS ON CYLINDER.

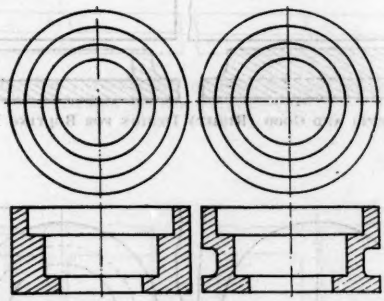


FIG 10—EXAMPLE WHERE GREATER STRENGTH IS OBTAINED BY LIGHTENING CASTING, AS SHOWN IN DESIGN AT RIGHT. ORIGINAL DESIGN AT LEFT.

Attention is directed especially to the coring out of the interior of the large flange visible in the foreground of the photograph. Figures 14, 15, 16 and 17 give details of this casting or examples that would be applicable to such a design.

37. In Figure 14 note the care taken to avoid inequalities of section on the projections shown.

38. Figure 15 shows a large rectangular flange, in the center of the picture, which has been carefully cored out below. Note also the easy rolling ribs and the rounded ridge extending the width of the photograph. This ridge is the outer and top surface of a key-way on the interior of the casting.

39. Figure 16 shows a round flange in the center of the picture, with a small boss on its right.

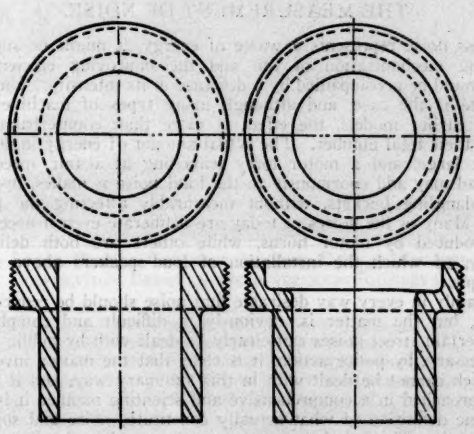


FIG. 11—POOR (LEFT) AND GOOD DESIGN (RIGHT) OF THREADED COLLAR.

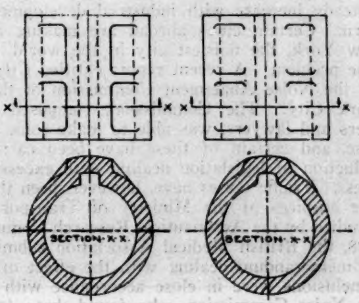


FIG. 12—POOR (LEFT) AND GOOD DESIGN (RIGHT) OF KEY-WAY.

40. The last photograph, Figure 17, is a close view of a large bolting pad, and attention is particularly called to the coring out of this projection and to the fact that sharp corners were avoided in accomplishing this modification.

41. The principles described have been applied in practical production with marked success, and it is hoped that the future will bring even more cooperation between the designer and the producer of castings so that alloys will not be selected solely for their mechanical properties, and that sounder and more serviceable castings will be the inevitable result.—Paper presented at Annual Meeting of American Foundrymen's Association.

THE MEASUREMENT OF NOISE.

As useless noise represents a waste of energy, it might be supposed that the growing mechanization of life and the improving efficiency of our machines would be accompanied by a decrease in its intensity. Unfortunately, the opposite is the case, and although many types of machine make less noise than earlier models, the effect is more than counterbalanced by an increase in their total number. The actual amount of energy in noise, *per se*, is also not large, and a motor lorry travelling at a fair speed along an average road may add enormously to the total noise it makes, by carrying a couple of dangling buckets, without measurably affecting the petrol consumption. Many of the noises of today are deliberate even if necessary, such as that produced by motor horns, while others are both deliberate and unnecessary, of which the installation of loud-speakers above shop doors is an example.

It is certainly in every way desirable that noise should be reduced as much as possible, but the matter is, obviously, a difficult and complicated one. Although certain street noises can clearly be dealt with by public regulations and, if necessary, by police action, it is clear that the matter involves many factors which cannot be dealt with in this summary way, and if the subject is to be approached in a comprehensive and scientific manner it is necessary to have some definition of what actually constitutes noise and some method of measuring it. This whole question was discussed in detail in a valuable paper read by Dr. G. W. C. Kaye, O.B.E., before the Royal Institution. Dr. Kaye began by dealing with the problem of noise as a concomitant of modern civilization. He pointed out that noise is common to all civilized nations, and its steady increase with industrial development is becoming a matter for concern. Certain cities abroad are making inquiries into the question, and New York, the noisiest city in the world, has in particular set to work on the problem. A recent report entitled *City Noise* was published in 1930 by the Noise Abatement Commission of the Department of Health, New York City. The Commission, composed of medical men, physicists, engineers and lawyers, was able to make some proposals for the abatement of noise, and certain of these have been carried out. In this country the introduction of legislation dealing with excessive noise has not made much progress. Traffic noises have, however, been the subject of conferences under the auspices of the Ministry of Transport, whilst aircraft noises are being studied by the Aeronautical Research Committee.

In October, 1928, the British Medical Association submitted to the Ministry of Health a memorandum dealing with the effect of noise on human beings. Their conclusions were in close accordance with those arrived at by the New York Noise Commission, who found that: (1) hearing is apt to be impaired in those exposed to constant loud noises; (2) noise inter-

FIG. 13—LARGE BRONZE CYLINDER OF GOOD DESIGN, SECURED THROUGH COOPERATION BETWEEN DESIGNER AND FOUNDRYMAN.

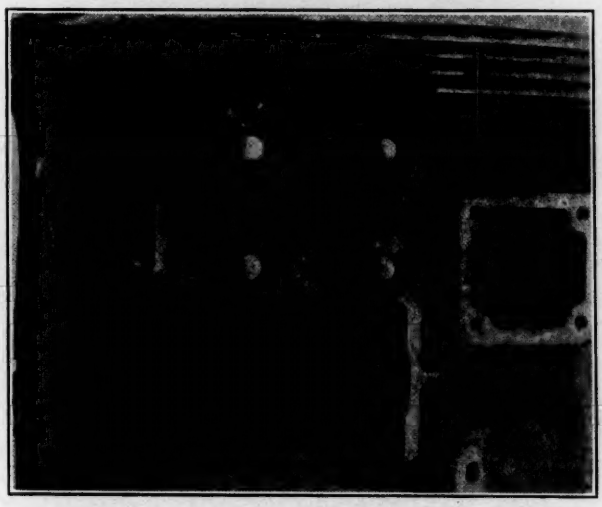


FIG. 14—DESIGN SHOWING CARE TAKEN TO AVOID INEQUALITIES IN SECTION OF CASTING.

FIG. 15—LARGE RECTANGULAR FLANGE CORED OUT UNDERNEATH TO AVOID INEQUALITIES OF SECTION. NOTE ALSO THE EASY ROLLING RIBS AND ROUNDED RIB OVER KEY-WAY.

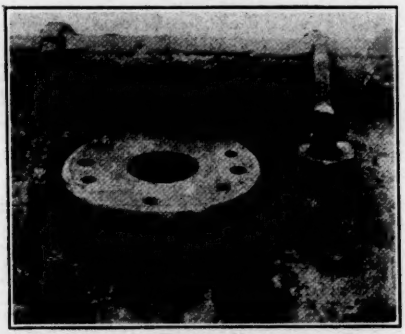


FIG. 16—GOOD DESIGN WITH ROUND FLANGE IN CENTER AND SMALL BOSS TO RIGHT.

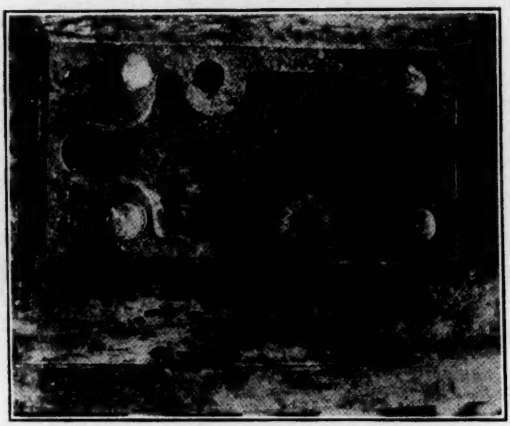


FIG. 17—LARGE BOLTING PAD AVOIDING SHARP CORNERS BY CORING OUT THE PAD.

feres seriously with the efficiency of the worker—it lessens attention and makes concentration on any task difficult; (3) in an attempt to overcome the effect of noise a great strain is put on the nervous system; (4) the normal development of infants and young children is hindered by constant loud noises. Reaction to noise is largely temperamental, and acuteness of hearing varies widely in different people. The question of noise tolerance seems, however, to be largely one not only of sensitivity, but of the "background" of noise that one is used to. If external sounds are completely excluded, adventitious noises are more trying. What is unconsciously desired is not so much the complete exclusion of noise as that the background shall be at an agreeably low level to which one is accustomed.

Dr. Kaye then turned to the question of the definition of noise. Most text-books on physics differentiate a musical sound from a noise by investing the latter with a complexity arising from complete irregularity of period, amplitude, and wave form.

In other words, a noise is to be regarded as a medley of notes of definite frequencies, the mixing being sufficiently random to obscure the musical quality of the individual notes. The average ear can perceive a range of frequencies from 20 cycles to 20,000 cycles per second, the upper limit declining with advancing years. In practice, attention is usually restricted to the ranges 50 to 5000 for speech and 35 to 7000 for music. As regards intensity, it is found that the ear is much more sensitive to notes of a medium pitch. Figure 1 shows the auditory sensation area for the average ear as determined by Fletcher and Wegel. The boundaries are constituted by two threshold curves. The range of audibility passes through a maximum (at about 1000 cycles per second) as the frequency is varied, and is greatly reduced towards each end of the musical scale. This maximum range corresponds to about a million millionfold variation in power, or a millionfold variation of acoustical pressure from about, say, 0.0005 dyne to 3000 dynes per square centimeter. We note also, in the case of very high and very low notes, the great intensity that is essential for audibility and how restricted is the range.

The matching and masking of sound form the basis, respectively, of the two aural methods of measuring the loudness of noise. As regards the matching of sounds, it is found that the average ear can recognize, under very favorable conditions, a 10 per cent difference of energy when two pure notes of medium loudness are sounded alternately without a break. Under ordinary conditions, however, the difference is of the order of 26 per cent for sounds of medium intensity and frequency. As it is a percentage rather than an additive increase which the ear associates with a change of loudness, the scale of sensation of loudness advances arithmetically, whilst the physical intensity increases geometrically. The physiological effect is, roughly, proportional to the logarithm of the energy producing the stimulus. Experiments carried out by Kingsbury indicated that, for frequencies between 700 cycles and 4000 cycles per second, the relation between loudness and intensity is independent of the frequency. Kingsbury's results are set out in Figure 2.

With reference to the masking of sounds, the masking of one pure note by another is usually measured by the extent to which the threshold of audibility of the masked note is raised. In general, a particular note is masked most readily of all by another of approximately the same frequency, and more readily by a note of lower than one of higher frequency, at any rate for loudness about and above speech level. The position is not so simple with loud complex sounds, such as noises, as the masking may be confused by other factors, such as the masking of the individual components, and the formation of subjective tones.

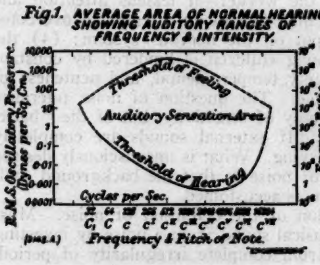


Fig.1. AVERAGE AREA OF NORMAL HEARING, SHOWING AUDITORY RANGES OF FREQUENCY & INTENSITY.

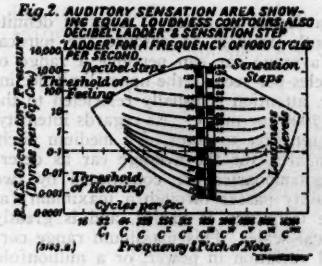
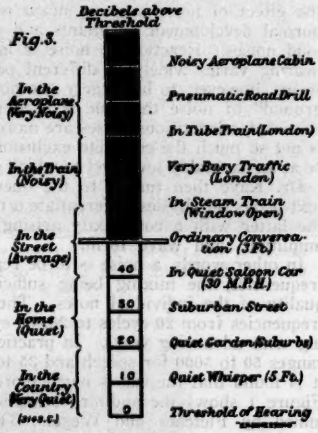


Fig.2. AUDITORY SENSATION AREA SHOWING EQUAL LOUDNESS CONTOURS, ALSO DECIBEL-LADDER & SENSATION STEP-LADDER FOR A FREQUENCY OF 1000 CYCLES PER SECOND.



LOUDNESS LEVELS OF COMMON NOISES

Fig.3.

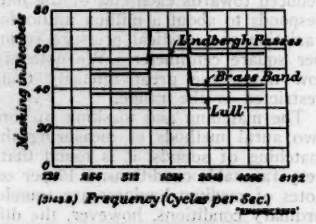


Fig.4. CROWD NOISES MEASURED 110 FEET AWAY.

In determining on a unit of noise, it is necessary to try to correlate aural loudness with physical intensity of energy; as already shown, while the sensation of loudness advances, as it were, by simple addition, the energy level increases on a scale which extends over almost astronomical magnitudes. This is a cumbersome relation, and it is clear that there will be a real convenience in adopting a scale of ratios of energy for our purpose. A similar need which arose in telephone engineering was met by the introduction of the "bel," a name chosen in honor of Alexander Graham Bell, the inventor of the telephone. One "bel" expresses a 10-fold increase of power or energy; in other words, two intensities in the ratio r: 1 differ by (log r) bels.* It has been generally agreed to adopt the bel for acoustic requirements also, or rather the "decibel" (db), since the bel is a little too large for the purpose. We thus have the following tabular relation:—

* Logarithms are to base 10.

Ratio (r) of Intensities.	No. of Decibels ($10 \log r$).
1	0
10	10
100	20
1000	30
10,000	40
.....	.....
10^{13}	130

It will be realized that this scale of decibels is in no sense physiological, but is based wholly on intensity as measured by physical methods. The scale has, however, the specific advantages, (1) of being a rough fit with the aural scale of loudness sensation, and (2) that experiment shows that the decibel, as above defined, corresponds approximately to the least perceptible change in loudness of a sound of medium loudness under average conditions. In actual fact, the loudness step in question is sometimes a little more and sometimes a little less than a decibel (ranging from 0.2 db. to 9 db.), according to the frequency and the location in the auditory sensation area.

It is now possible to set up a definition of the sensation level of a pure note of specified frequency in physical terms. The "degree" will be the decibel, the "zero," the threshold of audibility for that frequency, and the sensation level of a pure sound may accordingly be defined as the intensity in decibels above the threshold of audibility of that frequency. On this basis experiment shows that, for pure sounds of medium frequency, the range of audibility between the thresholds of hearing and feeling is covered by about 130 decibels. This figure is rather less for high and low frequencies. The position, however, is not so straightforward when one comes to the broader question of comparing the loudness of pure sounds of different frequencies. There is no simple relation of wide application between physical intensity and loudness. Two pure sounds of different frequencies do not, in general, produce the same loudness sensation, even if their physical intensities are equal, or their physical intensities bear the same ratio to their respective threshold values, *i.e.*, if the sounds have equal sensation levels. Further, if the intensity levels of two pure sounds of different frequencies which appear equally loud are increased by the same amount, the sounds will not, in general, remain equally loud.

It follows that, in the case of sounds of different or mixed frequencies, neither the physical intensity level nor the sensation level can be used as a measure of loudness. In the circumstances, an arbitrary scale is necessary as a practical standard, and a suitable one for the purpose is the sensation scale of a pure note having a frequency in the region of 1000 cycles per second. Then the loudness of any sound, whether pure, or a mixture such as noise, is defined as the sensation level (expressed in decibels above the threshold value) of the standard note which appears equally loud to the ear. It may be mentioned that the zero or threshold value of a 1000-cycle scale corresponds to a pressure of about a millidyne per square centimeter. It should be reiterated that such a standard scale is quite arbitrary and that equal increments on the scale do not, in general, approximate very closely to an equal number of sensation steps. For instance, in the case of the 1000-cycle scale, the step from 0 to 10 decibels corresponds roughly to about one perceptible gradation of loudness under average conditions, the step from 50 to 60 contains about 10 gradations, and the step from 100 to 110 about 15. In other words, one may ascend the range of physical intensity lying between the two thresholds, either by the decibel ladder with equally spaced rungs, or by the sensation step ladder with rungs first widely spaced and afterwards

more narrowly though more evenly spaced, as shown in Figure 2. It may be added that, in adopting a practical scale of loudness, some latitude is possible in the choice of the standard frequency, in view of the fact that, for medium frequencies above 700 cycles per second, there is a constant relation between loudness and sensation level. For example, the standard frequency chosen for much of the work at the National Physical Laboratory is 800 cycles, while in the United States 1000 cycles has been largely used.

The masking effect of a background of noise on the ease of conversation is one of primary interest. Conversation begins to be difficult when the background of noise reaches 70 or 80 db., whilst at 90 db. conversation, even by shouting, is impossible. Some 60 per cent of the energy of the human voice lies in frequencies below 500 cycles per second, and about 85 per cent below 1000 cycles. The intelligibility of speech rests largely on the high-frequency consonants—say above 1000 cycles, rather than on the low-frequency vowels (say 120 to 240 cycles). Experiments have shown that conversation in an inclosure, such as an aeroplane cabin, is facilitated if high-pitched noises are excluded. Fortunately, such notes can be more readily excluded than low notes; furthermore, they can be more readily absorbed by covering the walls with an absorbent.

The amount of energy of average speech is too small to adopt the standard method of measuring it by absorbing the sound waves in a suitable material and measuring the amount of heat generated; for example, the average speech power of the conversational voice is about 10 microwatts. This value rises to 1000 for a shouting voice, and falls to 0.001 for the softest whisper. A football match crowd of 100,000 talking continuously and rather loudly would provide sufficient power to light a small electric lamp. Probably the highest power developed is on ships' sirens, where measurements have shown a power-level of about 6 microwatts per square centimeter at a distance of 115 feet, so that the total energy would appear to be about ⅓-horsepower. In the case of a fog-siren an acoustical output of 1.7 horsepower has been measured. For sounds of ordinary magnitude we must turn to some other property of the sound waves, and for purposes of absolute measurement one of the most convenient is the Rayleigh disc which measures air or particle velocity.

The Rayleigh disc consists of a thin circular disc hanging vertically from its edge by a torsion thread, the disc being of small diameter compared with the wave length of the sound to be measured. Such a disc when placed in the sound field experiences a couple which tends to set the disc broadside on to the direction of the waves. This turning couple is independent of the frequency. We are thus enabled to calculate the particle velocity in the sound wave, provided we measure the deflection of the disc and the torsional constants of the system. In practice, discs of thin silvered glass are often used, suspended by quartz fibres some 15 centimeters long, and about 0.005 centimeters diameter. The Rayleigh disc is a fragile instrument, and more robust instruments are essential for practical purposes. Thus, electrical microphones, preferably of the non-resonant type, are generally used. These translate acoustical oscillations into electrical oscillations which are then passed to a valve amplifier. The condenser microphone is usually employed; it consists essentially of a metal diaphragm tightly stretched and mounted parallel to a metal plate with a gap of about 1/1000-inch. The resonant frequency is usually above the normal frequencies to be measured. A potential difference of 200 volts is applied through a high resistance. Sound waves incident on the diaphragm cause it to vibrate, resulting in variations of the condenser capacity, which, in turn, produces across the series resistance an alternating e.m.f. which can be amplified and measured. The condenser microphone for normal use in free air is calibrated with the Rayleigh disc.

The practical measurement of noise usually comprises one or more of the following operations: (1) The physical measurement of the "overall" power or energy-content of the noise, the result being expressible in absolute units (*e.g.* dynes or microwatts per square centimeter). (2) The physical analysis of the noise into its spectrum of frequency components (cycles per second). This is often most illuminating in tracking down the sources of individual components, particularly of machinery noises. (3) The physical determination of the wave form of the noise, though this is often difficult to interpret, and to utilize quantitatively, particularly if the noise is aperiodic. (4) The aural measurement in some accepted unit of the loudness of the noise, or in other words, the evaluation of the "noisiness" as perceived by the ear, which is the physiological arbiter of noise.

The question of the practical analysis of noises which are reasonably periodic, Dr. Kaye continued, has been much facilitated by the introduction of search-tone methods, which, in general, enable sound to be more conveniently analyzed, and with higher selectivity over a wide frequency range than is possible by the method of tuned circuits. When search-tone methods are employed, the noise to be analyzed is received in a microphone, the current of which is amplified and "mixed" in a valve-rectifier or modulator with that of a pure search tone of constant intensity and variable known frequency from a heterodyne oscillator. As a consequence, the modulated current contains not only the search tone, but also the summation and difference tones formed from the search tone and the various individual constituents of the noise. For example, if the frequency of the search tone is S, and that of a particular constituent of the sound is C, the frequencies of the summation and difference tone so formed will be $(S + C)$ and $(S - C)$, respectively. One way of revealing the existence of either of these tones is to apply the modulated current to a highly selective mechanical resonator, such as a steel bar, capable of vibrating longitudinally. Then, as the search frequency is varied continuously, the bar will begin to resonate whenever either $(S + C)$ or $(S - C)$ becomes equal to the natural frequency of the bar. As we know S, we can evaluate C, and further the degree of response of the bar, which is observed by suitable means, will give us a measure of the energy in the constituent in question. In practice, S may range from, say, 11,000 cycles to 16,000 cycles per second, while the natural frequency of the bar may well be of the order of 16,000 cycles per second for the summation-tone method, and 11,000 for the difference-tone method.

Grützmacher uses, instead of a mechanical resonator, a low-pass filter arranged to transmit frequencies less than about 30 cycles per second. It follows that, in the great majority of cases, both the search tone and all the summation tones are ruled out, and only those difference tones with frequencies less than 30 cycles per second will pass the filter and be recorded by an appropriate amplifier and detector. Thus, as the search frequency is continuously varied, the detector will only respond when the frequency is within 30 cycles per second of that of a constituent tone of the noise. The magnitude of the detector reading can be made to afford a measure of the intensity of the component in question. In the apparatus at the National Physical Laboratory, the frequency of the search tone is varied from, say, 30 cycles to 10,000 cycles per second by the rotation of an air condenser through 180 degrees.

As we have already seen, if we are provided with a standard pure note of medium frequency (above 700 cycles per second) the loudness of which is variable at will over a range which has been calibrated by physical means, then we can evaluate by aural matching or equality the loudness of any other pure note or, in general, of any complex note. Alternatively,

measurements may be made of the loudness of the standard note which is just masked or drowned by the sound to be measured.

Dr. Kaye was led, in 1929, to make some rough experiments on noise measurement by the aid of a flexed steel-strip "clicker," such as is sometimes used by lecturers. The note of the clicker is high pitched and surprisingly penetrating, and can be heard in quiet surroundings nearly 1000 feet away. The masking range of audibility was determined under a variety of conditions. In an aeroplane cabin or near a pneumatic road breaker or a riveter, the range shrinks to 2 feet or 3 feet, while near an aeroplane engine the range is only a few inches. On the somewhat doubtful assumption of the inverse-square law, the clicker confirmed the fact that the interior of a tube train (75 db. to 80 db.) is appreciably louder than that of an express train travelling at about 60 miles per hour—even in the corridor, with some of the windows open (70 db.). It is, of course, common knowledge that it is difficult to converse and listen in a tube train, but not difficult in an ordinary train with the windows closed, particularly in a first-class carriage, with its more generous upholstery. The cabin of an aeroplane in a cross-channel flight was found to be at least 1000 times (30 db.) noisier than an express train, although the plywood cabin walls cut down the noise of the engine 100 fold (20 db.). The preference exercised by knowledgeable passengers for seats in the rear of the cabin rather than in the region of the side propellers was confirmed, there being some 10 decibels difference. It was found that the customary practice of aeroplane passengers to plug their ears with cotton wool resulted in a reduction of the noise experienced by about 10 decibels.

A very convenient and portable means of measuring noise has been suggested and used by Davis at the National Physical Laboratory. A tuning fork is struck in some convenient standard manner—against the heel of the boot will do quite well—and no unusual care is necessary. The fork is then held with the flat of the prong towards the opening of the ear, and as close as possible without actually touching. The time of striking the fork is noted, and the interval of time is observed until the loudness falls to the level of the surrounding noise. If desired, the time interval before the note of the fork is masked by the noise can also be measured. The rate of decay of the fork is calibrated in decibels by a buzzer or other type of audiometer. As the decay of the loudness of a fork is practically logarithmic, the calibration curve of decibels against time is roughly linear. Readings are facilitated in practice if, as the fork is approaching the matching value, it is moved to and from the ear, so that its sound is alternately louder and softer than that of the noise. A particular fork used by Davis had a frequency of 640 cycles per second. Its loudness when it was struck was about 90 decibels, and the rate of decay was about 1½ decibels per second. Noises as high as 110 decibels were measured by means of masking observations. Davis has used the method for determining the loudness of a variety of noises over the range of hearing, and obtained results which are in good agreement with audiometer measurements by American observers.

Some simple illustrations of the measurement of every-day noises may be of service. The loudness of speech ranges between about 40 decibels and 60 decibels, an ordinary conversational tone being about 50 decibels. If, however, the lips of an average speaker are within ½ inch of the ear of a person with normal hearing, the latter will receive the speech at a level of about 100 decibels. An average motor horn sounded about 20 feet away, illustrates a loudness of about 80 decibels. Twins crying together are only 3 decibels louder than one crying alone. Another way of increasing a noise by 3 decibels is for the observer to move 30 per cent nearer when in the open air. If he moves another 20 per cent nearer, thus halving the distance,

the total gain will be 6 decibels. Fig 3 shows Dr. Kaye's "noise thermometer" of common noises ranging up to 100 decibels—a level which is unlikely to be exceeded in every-day experience. Some broad general groupings of noises are indicated on the left. Above 50 decibels the scale is differentiated as containing, in general, those noises which Dr. Kaye thinks we should do well to endeavor to moderate, as far as it may be practicable to do so, and we can, perhaps, regard this figure as a kind of "temperate" level on our noise thermometer.

A comparison between the loudness levels of traffic noises in London and New York, shows that for busy traffic the average decibels above the threshold value is 70 and 75, respectively, whilst for a quiet street, it is 50 and 60. It is stated by the New York Commission on Noise, that certain street corners are normally noisier than anywhere so far discovered in the world; as an example, the corner of Sixth Avenue and Thirty-fourth Street, where there are three main streets, three tram-car lines, a double track line of elevated railway, and the Subway (underground railway). Fig. 4 shows masking measurements of the crowd noises on the occasion of Lindbergh's arrival in New York after his Atlantic flight. The observers were on the fifth floor about 110 feet from the street. The masking effect of the noise was sufficient to mask the sound of a brass band not many yards distant. Among the contributory factors to traffic noise are motor horns. The New York Commission arrived at the conclusion that horns with sound levels in excess of 90 decibels, when heard 23 feet away, are unnecessary and objectionable. Complaints of stridency are unlikely to arise when fundamental frequencies lie between 200 cycles and 300 cycles, when the overtones are all harmonics of the fundamental and share the energy equally, and where there is an absence of strong high frequencies.

As regards trains, the noise level of express and suburban trains in England and America are similar. With reference to underground railways, the New York Subway stands in a class apart for noise; our tubes appear to be at least 10 decibels quieter. Among the loudest noises are those of riveting, pneumatic road drilling, steamship sirens and printing presses. Less common noises are those of lions and the Niagara Falls, which roar equally loudly (85 decibels). But the worst offender of all is the aeroplane engine at close quarters (110 decibels). The noise in the cabins of aeroplanes in flight ranges between 80 decibels and 110 decibels. There are, however, Dr. Kaye suggests, good prospects that the noise will be reduced (possibly to that of a railway train) by using propellers with lower tip speeds, effective silencers on the exhausts, reducing valve clatter, and constructing cabins with double walls.

There are two main practical methods for insulating an enclosure against air-borne noises:—(*a*) by using single non-porous rigid walls or partitions; and (*b*) by using multiple partitions as independent as possible and separated by air or some kind of loose filling.

In the case of a single rigid wall, the weight is the primary factor, the insulation value (in decibels) being proportional to the logarithm of the mass of the wall per square foot of area.

As a rough working rule, doubling the mass increases the insulation value by about 5 decibels, though resonance effects may spoil the relation. In the case of porous flexible materials, Knudson states that the insulating value is proportional to the mass of the wall per square foot of section, rather than to its logarithm. Often a combination of the porous flexible material and the rigid dense partition is advantageous.

As regards multiple partitions, they should be wholly free from cross ties, that is, completely isolated from one another, if the combination is to be any better (as it may be) than one single partition of the same overall thickness.

Whether or not a loose filling material should be sandwiched between the panels is a matter for experiment, as such a filler may or may not be beneficial. On the one hand, it may act as an absorbent and a damper of vibration, and on the other it may serve as a tie.

The question of protection from structure-borne noises is one to which it is hoped attention may be paid in the new Sound Laboratories at the National Physical Laboratory, as there is need for systematic experiment. Heterogeneity and discontinuity appear to be of value, and loose fillers may prevent the drumming of resonant panels or walls, which is often an accompaniment to such vibrations and may result in pronounced sound emission. In many houses the windows are the chief offenders in admitting external noises. Sound-proof construction would be greatly simplified if windows could be abolished. This, of course, is not to be contemplated, but we can at any rate use substantial windows of thick glass. Opening a window even a little will, of course, largely nullify the benefit of noise shielding and absorbing devices. Within limits, the amount of sound admitted by a crack or opening is proportional to its area. In the case of a door or window, affording, say, 30 decibels insulation, a crack with an area 1/1000 of that of the door would admit as much sound as passes through the door or window—Engineering, May 13, 1932.

BOOK REVIEW.

MOTORSHIPS OF THE WORLD. EDITED BY A. C. HARDY. PUBLISHED BY SAMPSON LOW, MARSTON & CO., LTD., LONDON. 10 x 7½ INCHES, 183 PP. PRICE 12/6 NET.

This volume, the third edition issued May, 1932, lists all the motor ships of the world in alphabetical order of names and gives for each ship the name of the owner, the names of the builders of ship and machinery, and the characteristics of the ships and of the machinery. There are also included separate lists of motor tankers, motor electric drive ships, and motor geared drive ships in addition to about one hundred photographs of typical motor ships.

—D.

ASSOCIATION NOTES.

In accordance with the By-laws, Captain Henry Williams (C. C.), U. S. Navy, was appointed a member of the Council vice Captain H. S. Howard (C. C.), U. S. Navy, who was detached in June from duty in the Bureau of Construction and Repair, and assigned to duty as Assistant Naval Attaché, London, taking with him the sincere thanks of the Society for his valued services on the Council since January, 1929.

MEMBERSHIP.

The following have joined the Society since the publication of the last previous JOURNAL:

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Cerpelli, A., Managing Director A. Cerpelli Co., Cassella Postal 103, La Spezia, Italy.

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